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Title: Simulation of engagement control in automotive dry-clutch and temperature field analysis through finite element model

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Keywords: dry-clutch; friction coefficient; temperature; finite element analysis; engagement driving

Corresponding Author: Dr. Mario Pisaturo, Ph.D.

Corresponding Author's Institution: University of Salerno

First Author: Mario Pisaturo, Ph.D.

Order of Authors: Mario Pisaturo, Ph.D.; Adolfo Senatore, prof.

Abstract: The tribological contact under sliding condition in the clutch facing surfaces during the engagement manoeuvre is strongly affected by heat transfer occurring in the system. The frictional forces acting on the contact surfaces produce mechanical energy losses which are converted in heat with ensuing temperature increase.
Reports about the temperature rise after repeated clutch engagements prove the occurrence of interface temperature peaks as high as 300°C.
Unfortunately, only few papers address their focus toward experiments and their outcomes about the influence of temperature and the other operating parameters on the frictional behaviour of the clutch facing materials.

In this paper, the Authors mainly explored the frictional behaviour modification for thermal level higher than 250-300°C, whose effect is a sharp decline of the friction coefficient related to the decomposition of the phenol resin of the facings. Moreover, this phenomenon induces not expected transition from dry friction to mixed dry-lubricated friction which explains the reasons of the friction coefficient drop. The temperature affects also the cushion spring load-deflection characteristic and the ensuing transmitted clutch torque.
Thus, an original frictional map has been implemented in a control algorithm to estimate the heat flux during vehicle launch and up-shift manoeuvres. The results of the longitudinal vehicle dynamics has been used in a FEA to predict the temperature field during repeated clutch engagement on the contact surfaces. The simulation results prove that during each engagement the interface temperature increases of 30-35°C. This means that after only few repeated clutch engagements the temperature field could reach values near the critical point of 300°C.
In such a way, this paper aims at providing useful references to control engineers in order to improve the dry-clutch transmissions performances.

Cover letter

Dear Editor,

as Authors of the paper "Simulation of engagement control in automotive dry-clutch and temperature field analysis through finite element model" submitted to this prestigious journal, we would like to briefly describe its main objectives.

Our research work could be classified in the area of automotive system and in particular it deals with the study of accurately estimate the temperature field in a dry-clutch during the engagement manoeuvres, start-up and gear-shift. To this aim, typical manoeuvres have been evaluated and implemented in a suitable control strategy based on the multiple Model Predictive Control (mMPC) designed to track reference trajectories both for the engine and clutch angular speed. Moreover, the controller takes into account the actuator and engine dynamics and a detailed dry-clutch behaviour.
Thus, an original frictional map has been implemented in a control algorithm to estimate the heat flux during vehicle launch and up-shift manoeuvres. The results of the longitudinal vehicle dynamics has been used in a FEA to predict the temperature field during repeated clutch engagement on the contact surfaces. In the proposed FE model the clutch facing material and the pressure plate have been modelled to get the temperature distribution due to the frictional energy. The simulation results prove that after repeated clutch engagements the temperature field reaches values near the critical point of 300°C.

The outcome of this analysis could become a key element for designers of automated clutches and control engineers to improve the dry-clutch transmissions performances.

Of course, we will be pleased to read the reviewers' comments in order to improve the paper quality and better fit the journal purpose.

Best Regards,
The Authors.

Suggested Reviewers

A. A. Yevtushenko, a.yevtushenko@pb.edu.pl, Faculty of Mechanical Engineering, Bialystok University of Technology

Michael Mitariu, mitariu@ipek.uni-karlsruhe.de, Institute of Product Development (IPEK), University of Karlsruhe (TH) Research University

Oday I. Abdullah, oday.abdullah@tu-harburg.de, System Technology and Mechanical Design Methodology, Hamburg University of Technology

HIGHLIGHTS

This paper focuses on the simulation of temperature field in automotive dry-clutch

FEA model has been used to predict the temperature rise at frictional interfaces

The roles of the clutch automatic control and of the vehicle dynamics are considered

The results could provide very useful issues to automotive transmission designers

Simulation of engagement control in automotive dry-clutch and temperature field analysis through finite element model

Mario Pisaturo*, Adolfo Senatore

Department of Industrial Engineering
University of Salerno
84084 Fisciano, Italy

Abstract

The tribological contact under sliding condition in the clutch facing surfaces during the engagement manoeuvre is strongly affected by heat transfer occurring in the system. The frictional forces acting on the contact surfaces produce mechanical energy losses which are converted in heat with ensuing temperature increase. Reports about the temperature rise after repeated clutch engagements prove the occurrence of interface temperature peaks as high as 300 °C. Unfortunately, only few papers address their focus toward experiments and their outcomes about the influence of temperature and the other operating parameters on the frictional behaviour of the clutch facing materials.

In this paper, the Authors mainly explored the frictional behaviour modification for thermal level higher than $250 - 300\,°C$, whose effect is a sharp decline of the friction coefficient related to the decomposition of the phenol resin of the facings. Moreover, this phenomenon induces not expected transition from dry friction to mixed dry-lubricated friction which explains the reasons of the friction coefficient drop. The temperature affects also the cushion spring load-deflection characteristic and the ensuing transmitted clutch torque. Thus, an original frictional map has been implemented in a control algorithm to estimate the heat flux during vehicle launch and up-shift

*Corresponding author
Email addresses: mpisaturo@unisa.it (Mario Pisaturo), a.senatore@unisa.it (Adolfo Senatore)

manoeuvres. The results of the longitudinal vehicle dynamics has been used in a FEA to predict the temperature field during repeated clutch engagement on the contact surfaces. The simulation results prove that during each engagement the interface temperature increases of $30 - 35\,^{\circ}\mathrm{C}$. This means that after only few repeated clutch engagements the temperature field could reach values near the critical point of $300\,^{\circ}\mathrm{C}$. In such a way, this paper aims at providing useful references to control engineers in order to improve the dry-clutch transmissions performances.

Keywords: dry-clutch, friction coefficient, temperature, finite element analysis, engagement driving

1. Introduction

Dry-clutch systems are used in a wide range of applications and one of the most important is the automotive industry. The aim of an automotive dry-clutch is to transfer the torque from the engine to the car wheels. In fact, when the clutch facings are in contact with the pressure plate on one side and the flywheel on the other side, by means of frictional phenomenon, the engine torque flows toward gearbox primary shaft and finally to the car wheels. In this application, as in the automotive brakes application, it is very important to have a high friction coefficient in order to maximize the effectiveness of the system. On the other hand, this means that during normal working conditions there is a noteworthy amount of kinetic energy converted into heat that ensue a temperature increase. The magnitude of temperature rise depends on thermal properties of the clutch facings [1]. This effect is amplified if the system is undergone to repeated load cycles like repeated clutch engagement or repeated breaks in a short time. These situations are very commonly if we consider that every day many drivers spend some hours in the traffic to go and come from work. The main problem is that the friction coefficient of the clutches and brakes linings drops off with the temperature reducing the functionality of the system. In fact, the heat generated may lead to a thermal decomposition of the frictional materials by producing fluids and gas, [2, 3, 4]. Another drawback of this phenomenon is the smell produced with the thermal decomposition of the phenolic resin perceived by the car passengers. In fact, it is one of the main claim to the assistance service. Reports about the temperature rise after repeated clutch engagements prove the occurrence of interface temperature peaks as high as $300\,^{\circ}\mathrm{C}$ [5] and [6].

In [7] the Authors have studied the temperature field in automotive dry-clutch during single and repeated engagements under two different hypothesis: uniform pressure and uniform wear. The same Authors have analysed the effect of groove pattern and groove size on the temperature field and thermal energy in [8, 9] and the surface temperature distribution of the dry friction clutch system by showing the effect of torque-time variation and dimensionless thickness of the pressure plate in [10].

In [11] the FEA analysis has been proposed to predict the facing wear as function of the contact pressure. In [12] a study to estimate the temperature distribution as function of the time and radius during the entire shifting process in dry clutch has been proposed.

Adamowicz and Grześ in [13] proposed a three-dimensional FEM to study and compare the temperature distributions caused by mutual sliding of two members of a disc brake system. They assumed that the rotor is subject to the non-axisymmetric thermal load to simulate realistic thermal behaviour of the brake action. They studied also the impact of convective mode of heat transfer on the thermal behaviour of a disc brake system during repetitive braking process with the constant velocity using fully three-dimensional finite element model [14].

Finally, in [15] a study on the thermal behaviour of the full and ventilated brake discs of the vehicles has been analysed.

Usually, in literature to estimate the heat flux in a dry-clutch during the engagement manoeuvres a linear profile for the sliding angular speed is assumed [7, 10]. In this work to accurately estimate the heat flux in a dry-clutch during the engagement manoeuvres, start-up and gear-shift, the vehicle longitudinal dynamics has been considered. In particular, typical manoeuvres have been evaluated and implemented in a suitable control strategy based on the multiple Model Predictive Control (mMPC) designed to track reference trajectories both for the engine and clutch angular speed [16]. Moreover, the controller takes into account the actuator and engine dynamics [17, 18] and a detailed dry-clutch behaviour [19]. To this aim, an original frictional map has been implemented in a control algorithm to estimate the heat flux during vehicle launch and up-shift manoeuvres. The results of the longitudinal vehicle dynamics has been used in a FEA to predict the temperature field during repeated clutch engagement on the contact surfaces. In the proposed FE model the clutch facing material and the pressure plate have been modelled to get the temperature distribution due to the frictional energy. The simulation results prove that after repeated clutch engagements the temperature field reaches values near the critical point of 300 °C. In such a way, this paper aims at providing useful references to control engineers in order to improve the dry-clutch transmissions performances.

2. Dry-clutch system

The aim of this section is to clarify the role of the main variables that explicitly influence the transmissibility of a dry-clutch system. The analysis of functional and structural links between the clutch engagement system and other driveline components is the first fundamental step for solving the modelling issues [20]. Recent studies have shown that the availability of phenomenological model of the torque transmissibility implemented in the TCU is fundamental to improve the performances of the automated dry-clutch. Until now the control strategies implemented in the TCUs take into account a simplified model of the friction phenomena and without considering the operation of clutch subsystems [21]. Indeed, these latter have a remarkable influence on the shifting quality of the gear shifts and consequently on the performances of the automated transmission. By describing a typical engagement process it has been illustrated how the diaphragm spring [22, 23, 24] and the cushion spring [25, 26] take part in the transmissibility characteristic. Even though the torque transmitted through a dry-clutch influences and is influenced by driveline components and diaphragm spring, the torque transmissibility proposed model shows that the two main elements to be considered are the cushion spring load and the dry friction phenomenon. In the next subsections, how these two elements enter separately into the proposed torque transmissibility model is shown. In such model the influence of the temperature [2, 5, 27, 28], of the slip speed and of the contact pressure [29, 30, 19] has been taken into account. Finally a novel torque transmissibility model of the automated dry-clutch such as AMT and DCT has been defined. In Figure 1 the dry-clutch architecture is showed.

2.1. Friction coefficient

The friction coefficient has a strong effect on the clutch torque characteristic as it is shown in the next paragraphs. For this reason a deep analysis on its variation during the engagement phase is fundamental to improve the performance of an actuated dry-clutch. The modelling of friction variation during the clutch engagement process has been studied by numerous authors.

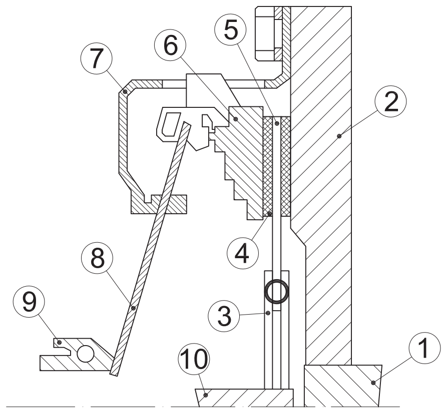

Figure 1: Dry-clutch architecture: 1 crank-shaft, 2 flywheel, 3 clutch disc, 4 clutch facings, 5 cushion spring, 6 pressure plate, 7 cover, 8 diaphragm spring, 9 throwout bearing, 10 primary shaft

In this section is explained how the temperature, the sliding speed and average contact pressure affect the friction coefficient. In [30, 19] a detailed analysis is reported therefore for the sake of brevity here are highlighted only the main results.

In Figure 2 the technical data sheet of a typical automotive clutch facings is showed [27]. In that graph the friction coefficient exhibits a smooth variation within the temperature range from 100 °C to 250 °C, whereas it begins to sharply decline above 250 °C. This typical behaviour is also shown in [2, 3]. This effect is related to the decomposition of the phenolic resin of the clutch facings at higher temperature. In fact, when the temperature is higher than 330 °C, a severe thermal decomposition produces fluids and gas emissions. This effect induces not expected transition phenomena from dry friction to lubricated friction. For this reason the friction coefficient drops [2, 3].

Instead, in Figure 3 is pointed out the relationship between the friction coefficient and the sliding speed for two different average contact pressures, [30]. It is evident that for both contact pressure level the friction coefficient tends to an asymptotic value at higher sliding speeds. The friction coefficient asymptotic value is higher for higher contact pressure. Moreover, the contact pressure has a nearly linear influence on the friction coefficient, according to the results in [30], achieved on a tribometer at room temperature, Figure 3.

Thus, the dependence of the friction coefficient on sliding speed, average

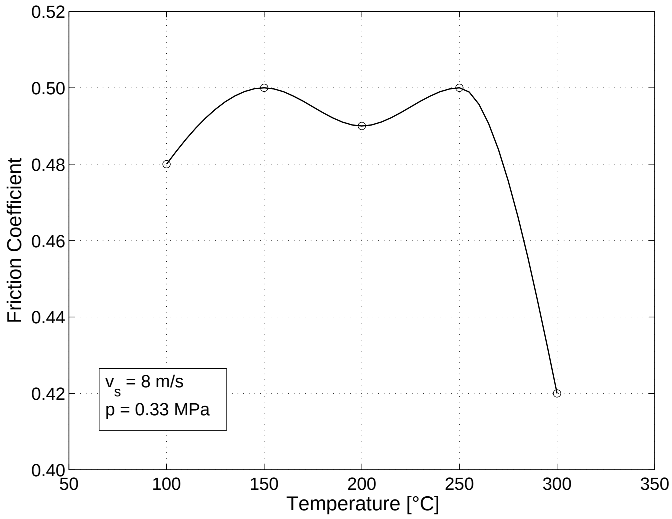

Figure 2: Friction coefficient vs. average facing temperature, [27]

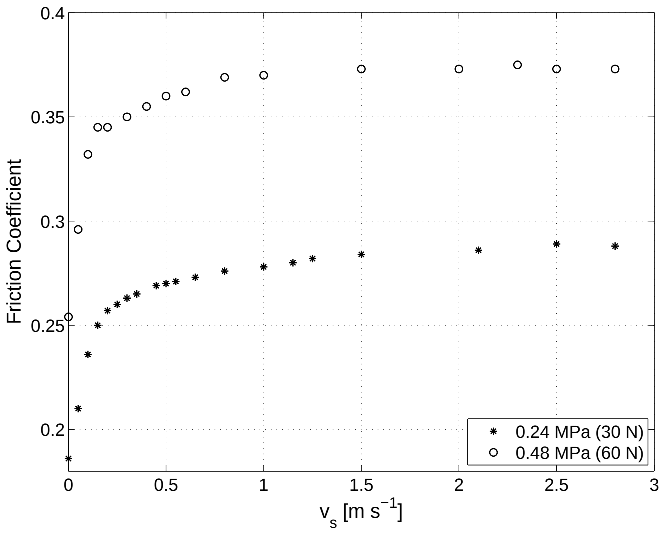

Figure 3: Friction coefficient vs. sliding speed at different contact pressure: 0.24 MPa, 0.48 MPa, [30]

contact pressure and facing temperature is given by the equations (1) and (2):

$$\mu_{\infty}\left(p, \theta_{cm}\right) = \alpha + \beta \frac{p}{p_0} + f\left(\theta_{cm}\right) \tag{1}$$

where α and β have been identified from the Figure 3 in the plateau region and the corresponding values are: $\alpha = -0.110$, $\beta = 0.110$; the reference pressure is $p_0 = 0.33\ MPa$, i.e. the same average contact pressure of the experiments in Figure 2, [27]. The function f is derived from the data in Figure 2.

The equation (2) is based on the experimental results in [30] for strictly positive slip speed:

$$\mu\left(v_s, p, \theta_{cm}\right) = \mu_{\infty}\left(p, \theta_{cm}\right) + \mu_{\Delta}\Big(\tanh\left(\frac{v_s}{v^*}\right) - 1\Big) \tag{2}$$

In equation (2), $v_s = R_m\left|\omega_e - \omega_c\right|$ is the sliding speed given by the product between the difference between engine and clutch angular speed $|\omega_e - \omega_c|$ and mean radius R_m. μ_{Δ} and v^* have been identified and the corresponding values are 0.09 and 0.5 m/s, respectively.

2.2. Cushion spring

The cushion spring is a thin steel disc placed between the clutch friction pads and is designed with different radial stiffness in order to ensure the desired smoothness of engagement [21]. When the cushion spring is completely compressed by the pressure plate we say that the clutch is *closed*, whereas when the pressure plate position is such that the cushion spring is not compressed we say that the clutch is *open*. We say that the clutch is in the slipping phase when is going form open to locked-up, i.e. the flywheel and the clutch disc have the same angular speed $\omega_e = \omega_c$. The temperature influences the cushion spring load-deflection characteristic [26] and this latter influences the clutch torque transmissibility. In [28] a detailed finite element analysis has been carried out in order to evaluate the influence of the temperature on the cushion spring characteristic, Figure 4.

2.3. Transmissibility model

By considering the results shown in the previous subsections on the cushion spring and on the friction coefficient it has been developed a more complex clutch torque transmissibility model reported in the equation (3), [19].

$$T_{fc}(y, \theta_{cs}, \theta_{cm}, v_s, p) = n\mu(v_s, p, \theta_{cm}) R_m F_{fc}(\delta_f(x_{pp}(y, \theta_{cs}), \theta_{cs})) \tag{3}$$

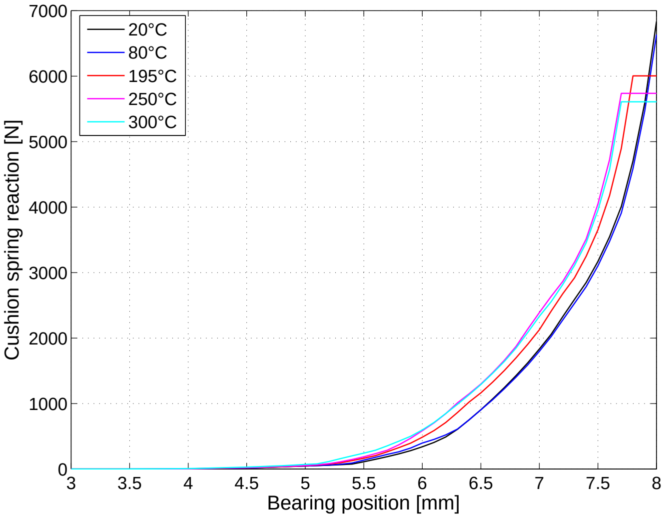

Figure 4: Cushion spring characteristic at increasing temperature levels, [28]

where T_{fc} is the torque transmitted by the clutch, $R_m = \frac{2}{3}\frac{R_e^3 - R_i^3}{R_e^2 - R_i^2}$ is a geometrical parameter obtained under the assumption of a uniform pressure, y is the throwout bearing position, θ_{cs} and θ_{cm} are the cushion spring and the clutch material temperature respectively, F_{fc} is the cushion spring reaction. This model has been implemented in a control algorithm in order to describe the behaviour of the dry-clutch architecture.

3. Heat flux estimation

As mentioned in the paragraph 1, in this work to accurately estimate the heat flux in a dry-clutch during the engagement manoeuvres, the vehicle longitudinal dynamics has been considered. In particular, typical manoeuvres have been evaluated and implemented in a suitable control strategy based on the multiple Model Predictive Control (mMPC) [16] which take into account the actuator and engine dynamics [17, 18] and a detailed dry-clutch behaviour [19]. For the sake of brevity the equations that describe the vehicle longitudinal dynamics are omitted but a detailed mathematical representation can be found in the cited literature. In Figure 5 the control scheme implemented in MATLAB/Simulink has been reported. The heat flux estimated with this approach has been used into a FEM model to calculate the temperature field in the dry-clutch components.

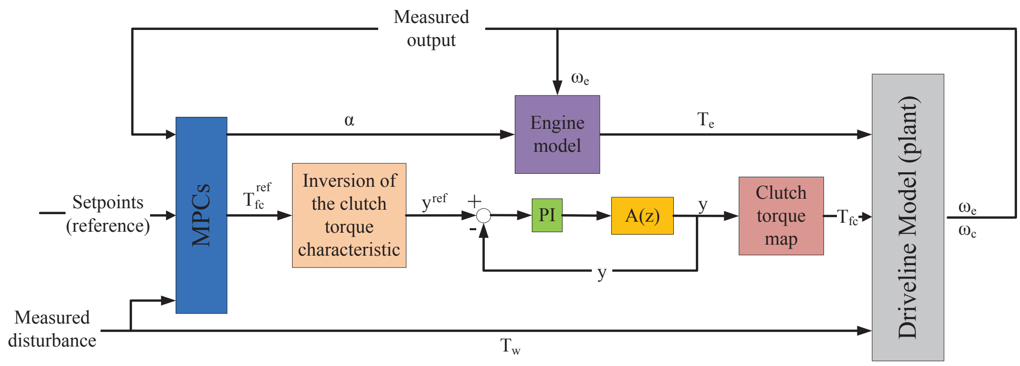

Figure 5: Control scheme

In particular has been simulated three different manoeuvres, both vehicle launches and gear-shift, to estimate the heat flux in each working condition. The aim is to evaluate how the thermal power Q varies with the time by using the outputs of the control algorithm, namely the angular sliding speed ω_{sl} and the frictional torque transmitted by the clutch during the manoeuvre T_{fc}. By assuming that all the frictional work is converted into heat, the thermal power during the clutch engagement is given by:

$$Q\,(t) = T_{fc}\,(t)\,\omega_{sl}\,(t) \tag{4}$$

Finally, by considering the contact surface equal to the interface geometrical surface the heat flux has been evaluated.

3.1. Manoeuvre 1

In this subsection the plots of the main results concerning a vehicle launch manoeuvre are reported. In Figure 6 the engine, clutch and sliding angular speed have been reported.

Instead, Figure 7 shows the frictional torque transmitted by the clutch during the engagement manoeuvre.

As explained previously the frictional work converted into heat, and consequently the thermal power, generated during the clutch engagement have been estimate by the equation (4) by using the values highlighted in the above figures. In Figure 8 the thermal power and the heat flux due to the simulated start-up manoeuvre have been plotted.

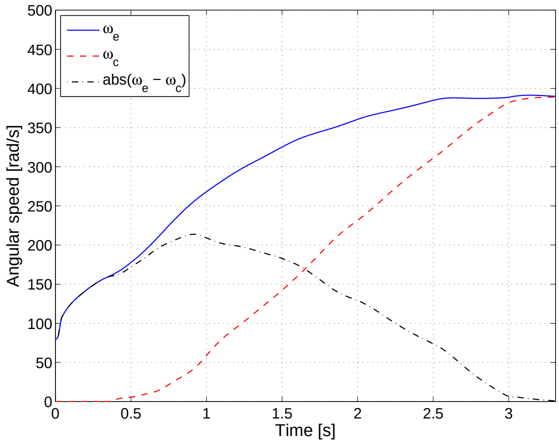

Figure 6: Manoeuvre 1 - Angular speeds

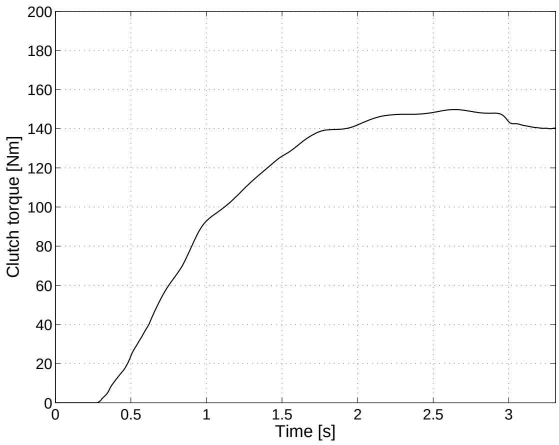

Figure 7: Manoeuvre 1 - Clutch torque

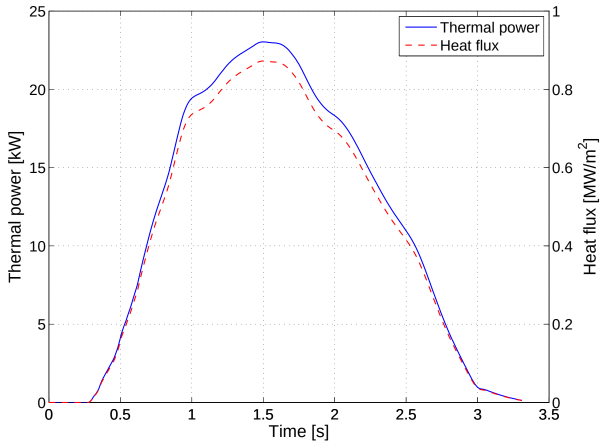

Figure 8: Manoeuvre 1 - Thermal power and heat flux

3.2. Manoeuvre 2

In this subsection another vehicle launch manoeuvre has been analysed in the same way explained above. In Figure 9 the engine, clutch and sliding angular speed have been reported. It is worth to note that in this case the engine angular speed increases suddenly in the first half second by resulting in a higher sliding speed and consequently in a higher thermal power generated during the manoeuvre respect to the previous one.

Instead, Figure 10 shows the frictional torque transmitted by the clutch during the engagement manoeuvre.

As explained previously the frictional work converted into heat, and consequently the thermal power, generated during the clutch engagement have been estimate by the equation (4) by using the values highlighted in the above figures. In Figure 11 the thermal power and the heat flux due to the simulated start-up manoeuvre have been pointed out.

3.3. Manoeuvre 3

Finally, a vehicle launch manoeuvre together to an up-shift $1^{st} - 2^{nd}$ has been considered in order to compare the thermal power generated during these two different working conditions. In Figure 12 the engine, clutch and sliding angular speed obtained in this case have been reported. It underlines that heat generated during the up-shift is negligible. In fact, during a gear-shift request the transmission control unit gives the command to open the

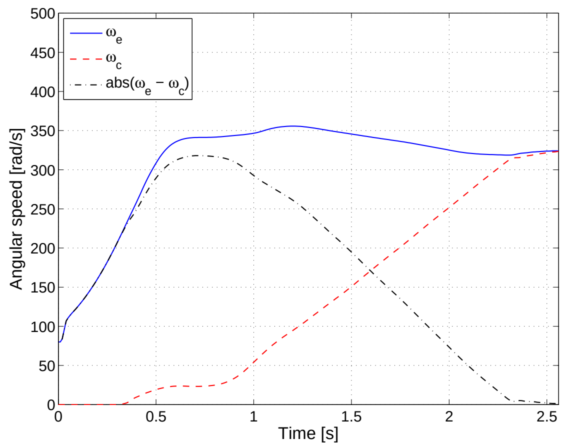

Figure 9: Manoeuvre 2 - Angular speeds

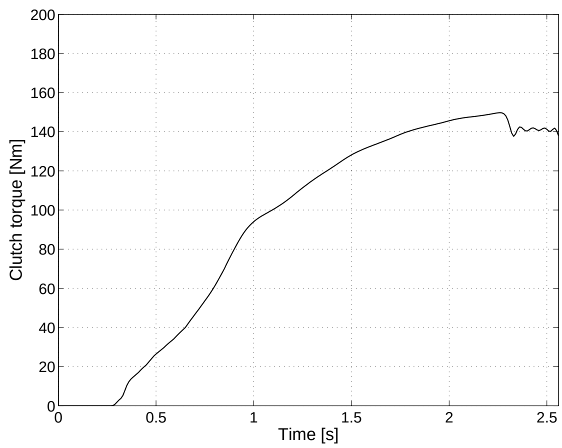

Figure 10: Manoeuvre 2 - Clutch torque

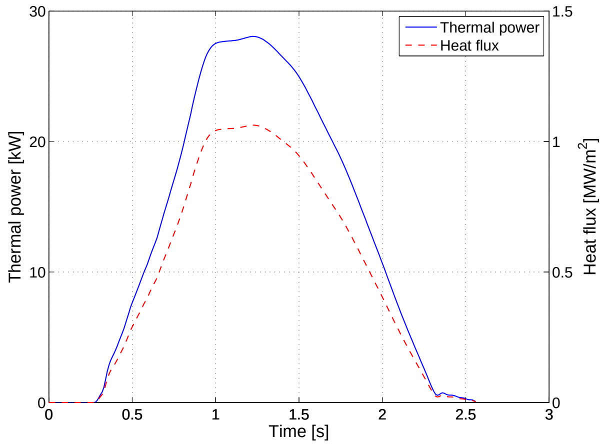

Figure 11: Manoeuvre 3 - Thermal power and heat flux

clutch as fast as possible and consequently the clutch torque go to zero. When the requested gear is engaged and the synchronization phase is completed the clutch speed disc moves quickly the speed value corresponding to the vehicle speed reported to the main-shaft through the new gear ratio [31].

Instead, Figure 13 shows the frictional torque transmitted by the clutch during the manoeuvre.

In Figure 14 the thermal power and the heat flux due to the simulated start-up manoeuvre have been plotted.

4. FE model

To appraise the interface temperature during the engagement manoeuvres a Finite Element Model (FEM) has been implemented by using a commercial software. In particular, by considering the axisymmetric of the system a 2D-model has been considered in the numerical simulations. In Figure 15 the clutch system which has been taken into account is reported. The values obtained from the simulations take into account the total amount of the heat generated during the manoeuvres, i.e. on both the contact pairs surfaces. But, in the proposed 2D-model only one contact pair is analysed. Thus, by considering the symmetry on the mid-plane the hypothesis that in each contact pair is generated the 50% of the total heat has been made. Under this latter consideration a thermal symmetry condition has been imposed on

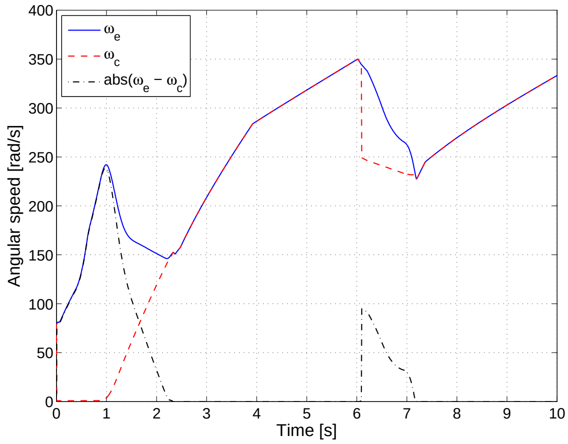

Figure 12: Manoeuvre 3 - Angular speeds

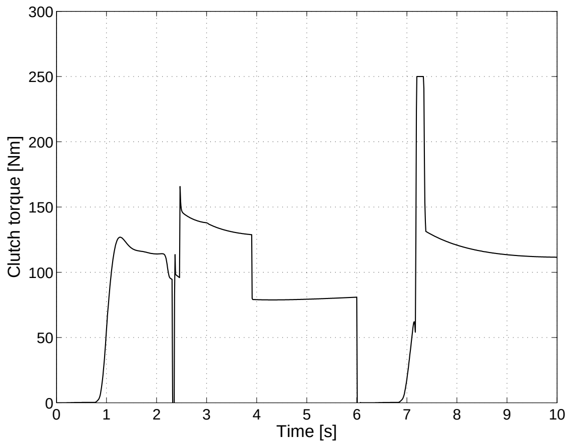

Figure 13: Manoeuvre 3 - Clutch torque

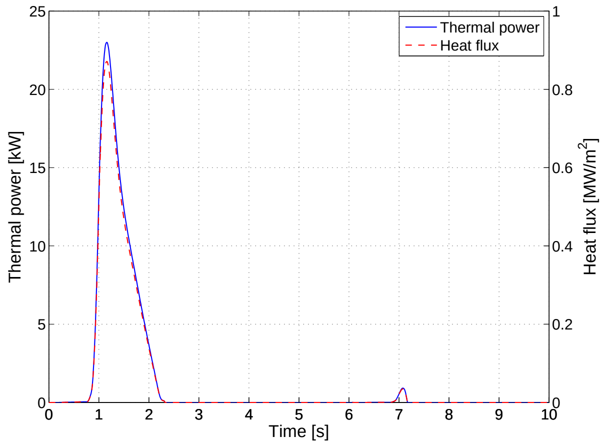

Figure 14: Manoeuvre 3 - Thermal power and heat flux

the mid-plane, Figure 15. Finally, on all the surfaces in contact with the air conductive losses, by using a conductive coefficient of 20 $Wm^{-2}K^{-1}$ and a bulk temperature of 298 K, have been imposed. So, only conduction and convection thermal mechanism has been considered in this paper. In fact, due to the short time and the relatively low temperatures reached during the engagement it is possible to neglect the radiant phenomenon.

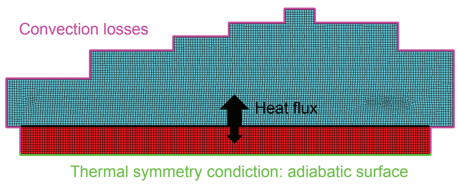

Figure 15: Finite element model

As mentioned before, a uniform contact pressure between the clutch disc and the pressure plate has been considered to calculate the frictional work converted into heat. This means that the heat flux is independent from the spatial distribution. Furthermore, it was assumed that the heat flows in a direction perpendicular to the contact surfaces. The heat partition ratio has been calculate with the Charron's formula recommended for brakes and clutches applications [32], eq. (5):

$$\gamma = \frac{\sqrt{K_1\rho_1C_1}}{\sqrt{K_1\rho_1C_1} + \sqrt{K_2\rho_2C_2}} \tag{5}$$

where K is the thermal conductivity, ρ the density and C the specific heat capacity. The subscribes 1 indicate the clutch disc and 2 the pressure plate respectively. The heat partition ratio has been calculate by considering the materials proprieties of both the bodies isotropic and independent from the temperature. In Table 1 the materials proprieties used for the numerical analysis are listed:

Table 1: Material proprieties

Parameters	**Pressure plate**	**clutch disc**
Density $[kgm^{-3}]$	7250	1540
Young's module $[GPa]$	150	3.68
Poisson ratio	0.28	0.25
Thermal conductivity $[Wm^{-1}K^{-1}]$	48	0.70
Specific heat capacity $[Jkg^{-1}K^{-1}]$	540	750
Linear expansion $[K^{-1}]$	10^{-5}	$16\ 10^{-6}$

5. Simulation Results

In this paragraph the results of the numerical simulation described before are reported. In particular, the goal of this paper is to estimate the temperature increase after repetitive engagements for each manoeuvre which could represent urban traffic scenario. To this aim each manoeuvre has been repeated in order to cover a window time of about 40 s. Details on the load cycle considered in each case are explained in the dedicated subsection.

5.1. Manoeuvre 1

In Figure 16 the interface temperature and the temperature on the cushion spring side versus time at different radii are plotted. The picture highlights also the single load cycle which has a first part that represents the slipping phase of 3.45 s and a second part of 5 s which represent the engaged phase. The same load cycle has been repeated 5 times to cover the window time considered. This picture shows how the interface temperature (solid lines) rises during the first part of the slipping phases (about $30 - 35$ K)

and it decreases during the engaged phases under the effect of the convective losses. The results highlight that the convective losses can not dissipate completely the thermal energy generated during the slipping phase. Moreover, after only five repeated engagements the interface temperature can attain about $430 - 450\ K$, i.e. about $160 - 180\,^{\circ}\mathrm{C}$, near the critical temperature value for clutch materials [6, 2]. The temperature on the cushion spring side (dashed lines) presents a behaviour smoother than the interface temperature due to the effect of the clutch material thickness. In any case, it is worth noting that after repeated engagements also on the cushion spring side it is possible to attain high temperature. This results in a change of the cushion spring load-deflection characteristic as showed in the section 2.

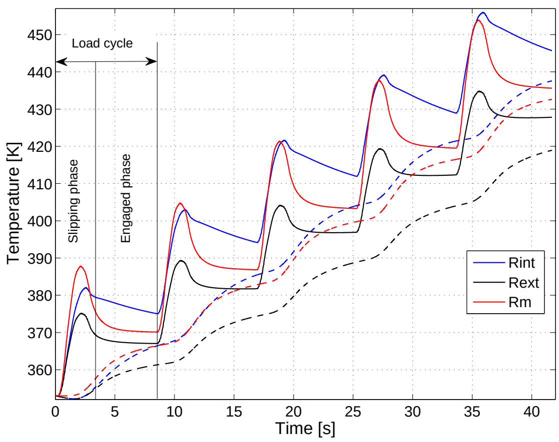

Figure 16: Manoeuvre 1 - Interface Temperatures (solid lines) and cushion spring side (dashed lines) vs. time after repeated engagements

The differences between the temperatures at different radii are due to the irregular geometry of the pressure plate which results in a non-uniform temperature distribution, Figure 17. Indeed, on the external radius the temperature increase is lower than on the internal and medium radii due to the irregular geometry of the pressure plate. Indeed, this latter presents in the internal zone a reduced thickness that results in a higher cumulated thermal energy during each load cycle that the convective losses can not dissipate completely.

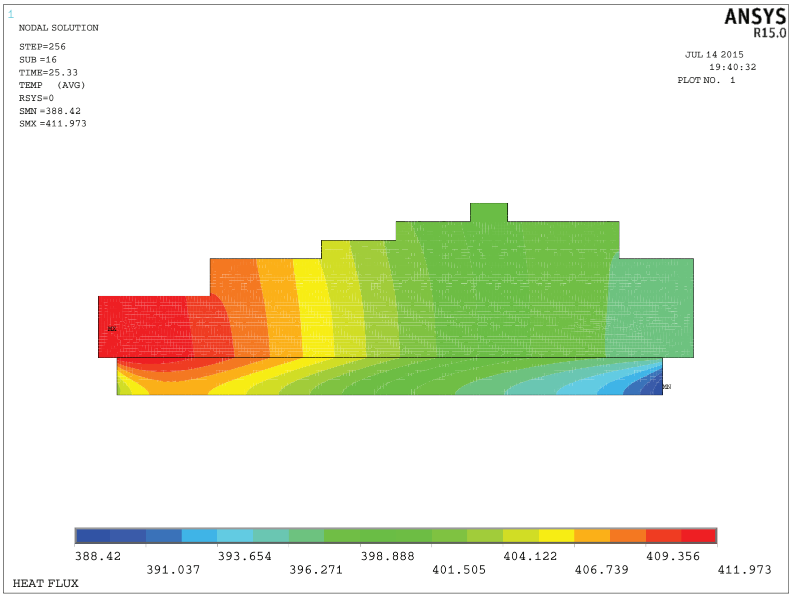

Figure 17: Manoeuvre 1 - Temperature distribution at time step 25.3 s

5.2. Manoeuvre 2

In Figure 18 the interface temperature and the temperature on the cushion spring side versus time at different radii are plotted for the second manoeuvre. In this case the slipping phase is of 2.70 s and the engaged phase is of 5 s. Also in this simulation the same load cycle has been repeated 5 times to cover the window time considered. As explained previously during this manoeuvre the frictional work converted into heat is higher than previous one, so in this case the interface temperature rises of about $30 - 40K$. The behaviour of the temperature on both sides of the clutch materials during the engaged phases is the same observed for the manoeuvre 1. Indeed, also these results highlight that the convective losses can not dissipate completely the thermal energy generated during the slipping phase. Moreover, after only five repeated engagements the interface temperature can attain a temperature close to the critical value for clutch materials.

5.3. Manoeuvre 3

Finally, the third manoeuvre represents a vehicle launch and an up-shift and as underlined above the heat generated during the up-shift is negligible.

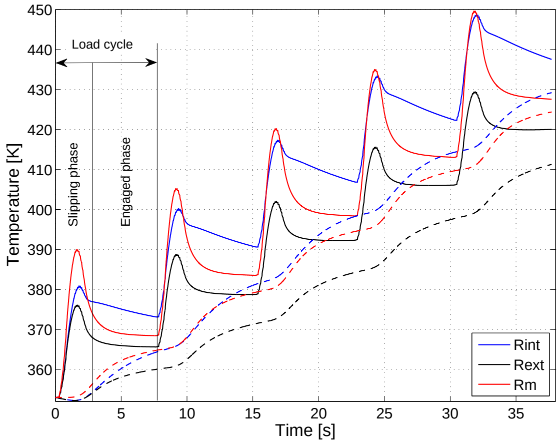

Figure 18: Manoeuvre 2 - Interface Temperatures (solid lines) and cushion spring side (dashed lines) vs. time after repeated engagements

For these reasons during a gear-shift manoeuvre the generated heat is negligible and in the load cycle reported in Figure 19 the temperature increase due to the up-shift is piratically zero. For this manoeuvre the frictional work converted into heat is lower than the previous cases and concentrated in a short time. This results in a temperature increase lower than the manoeuvres reported previously. Moreover, as highlighted in Figure 19 the interface temperature and the temperature on the cushion spring side versus time at different radii have the same trend discussed above.

6. Concluding Remarks

By concluding, in this paper three typical vehicle launch manoeuvres have been analysed to estimate the heat flux due to the friction phenomenon in a dry-clutch assembly. These results have been used to set up an axisymmetric finite element model of a clutch facing in contact with the pressure plate. The aim of the FE analysis is to calculate the temperature increase after repeated clutch engagements. The simulation results have showed that during each slipping phase the temperature increase can reach about $30 - 35\ K$. On the other hand, during the engaged phase the convective losses cannot dissipate completely this thermal energy. Moreover, after only 5 repeated engagements the interface temperature can amount to about $430 - 450\ K$. Namely, the thermal load in these working conditions could lead to critical temperature

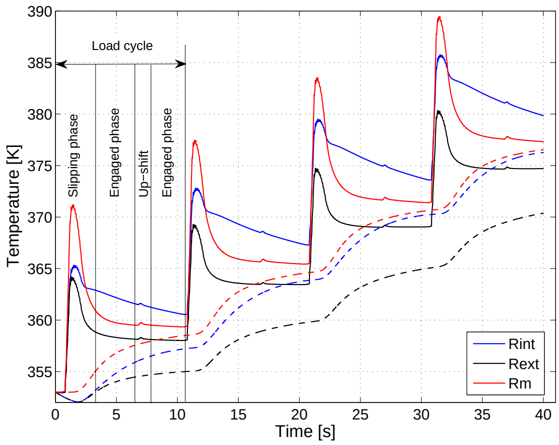

Figure 19: Manoeuvre 2 - Interface Temperatures (solid lines) and cushion spring side (dashed lines) vs. time after repeated engagements

values after few repeated engagements. This means that the clutch facings could suffer permanent damages together with smell perceived by the car passengers and a reduction of the performances.

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Fig. 1

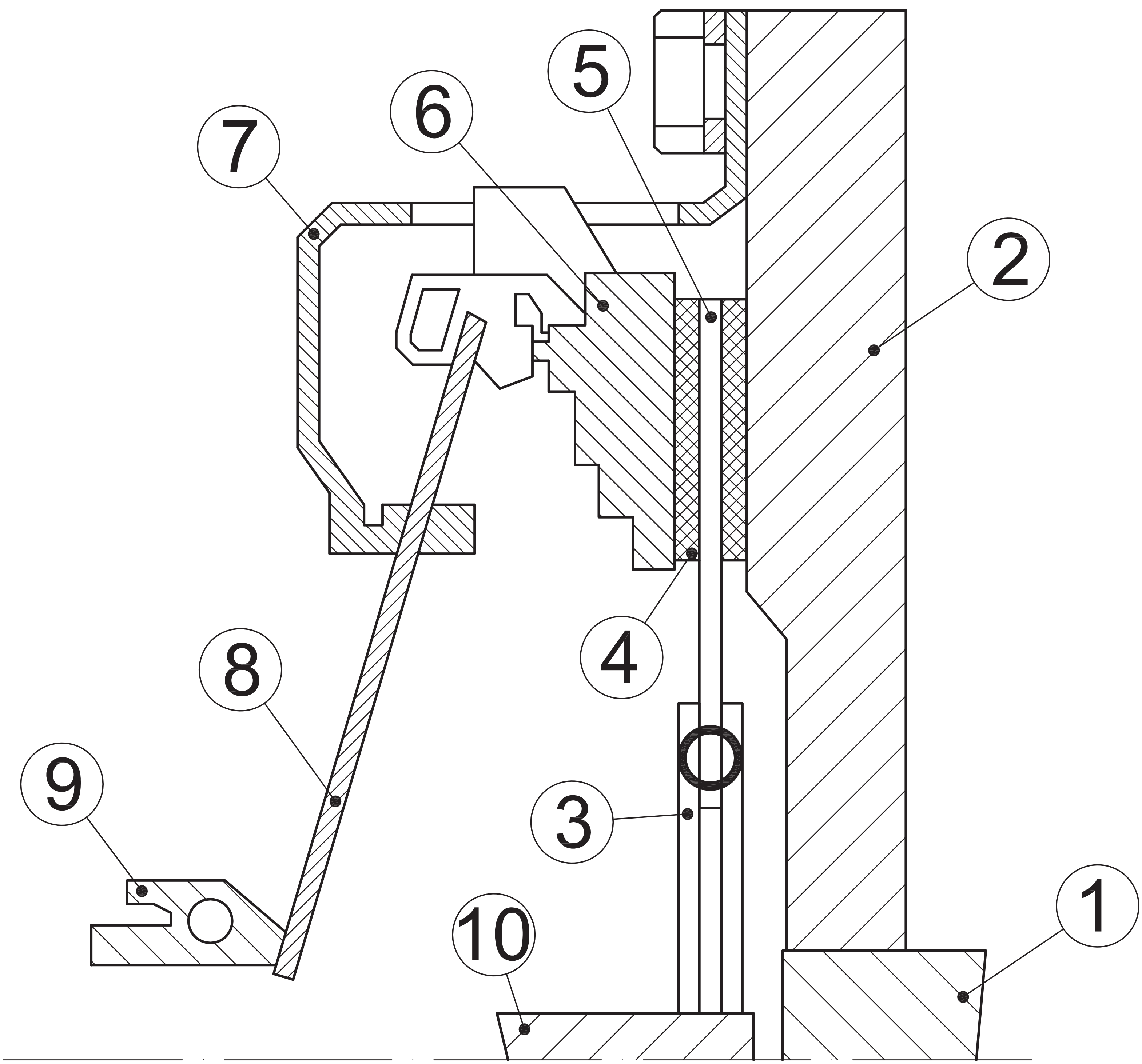


Fig. 2

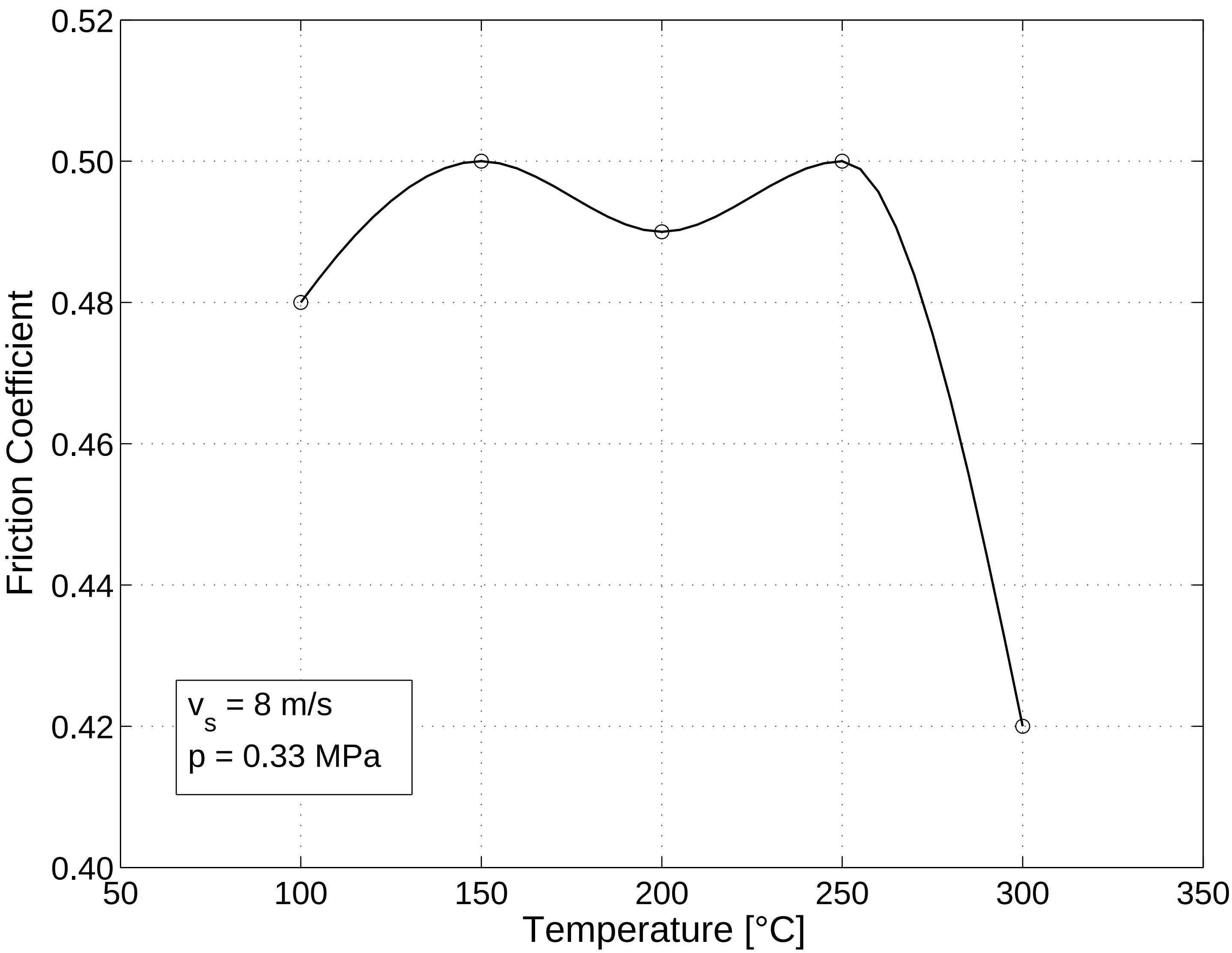

0.52
0.50
0.48
0.46
0.44
0.42
0.40
Friction Coefficient
50
100
150
200
250
300
350
Temperature [°C]
v_s = 8 m/s
p = 0.33 MPa

Fig. 3

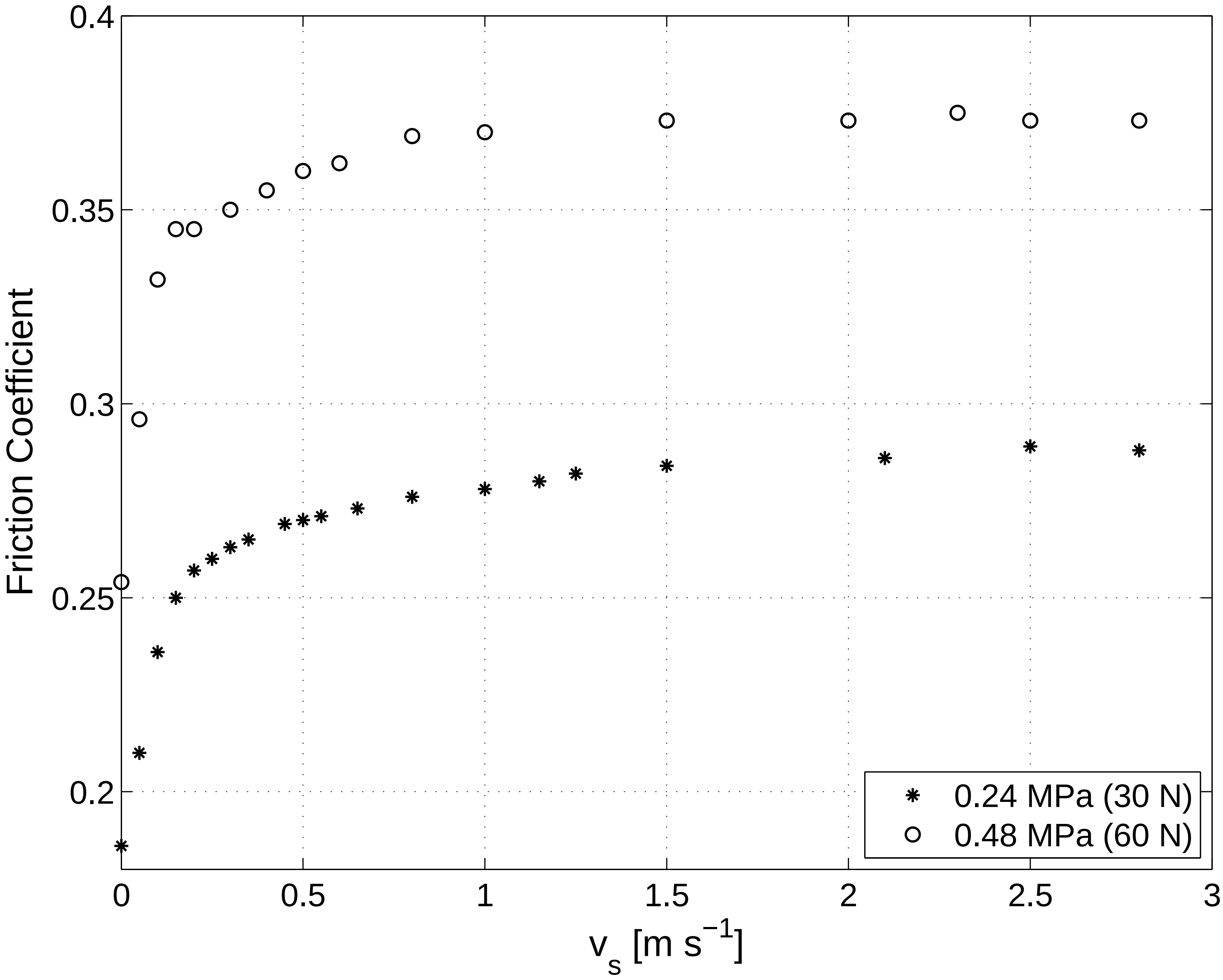
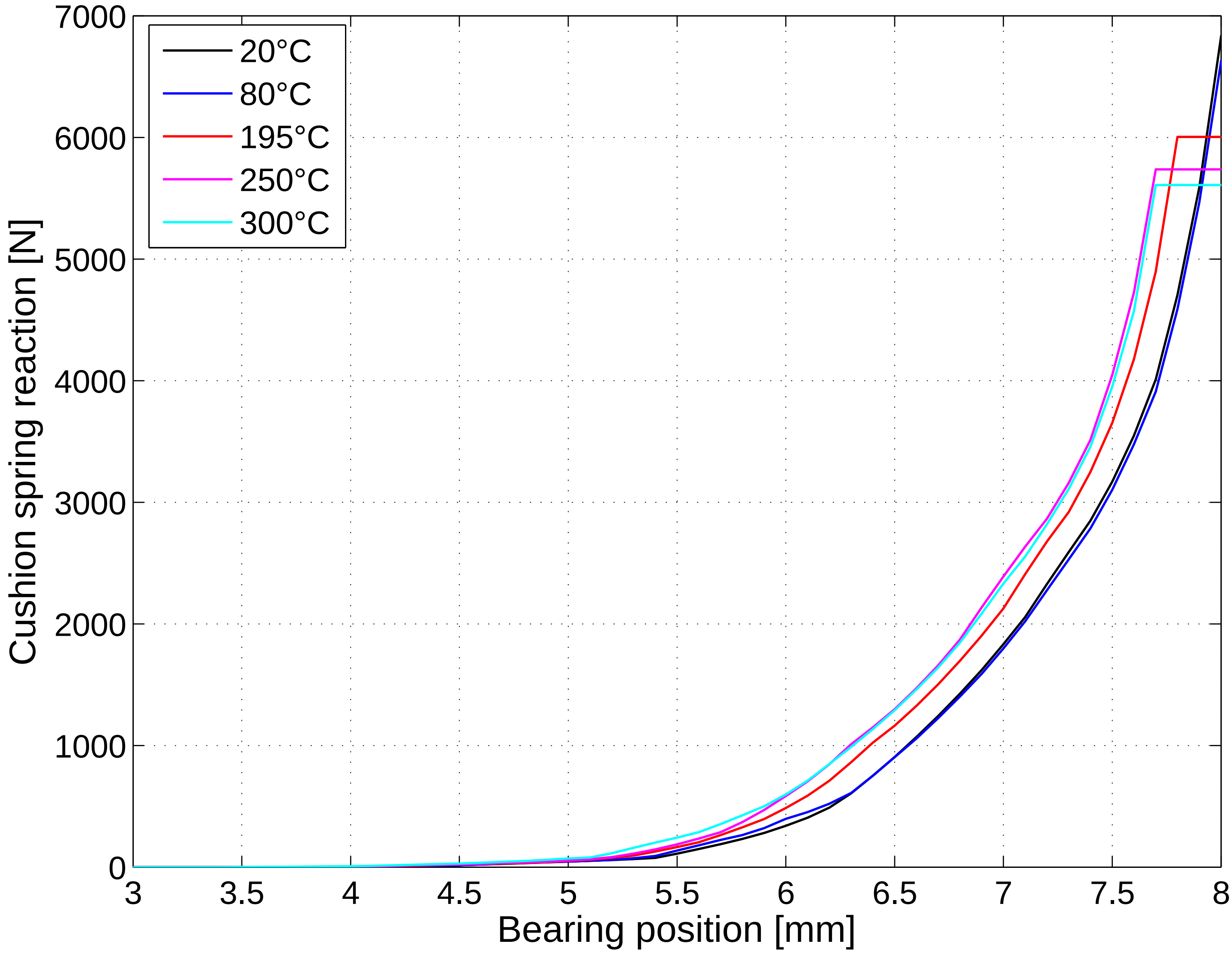
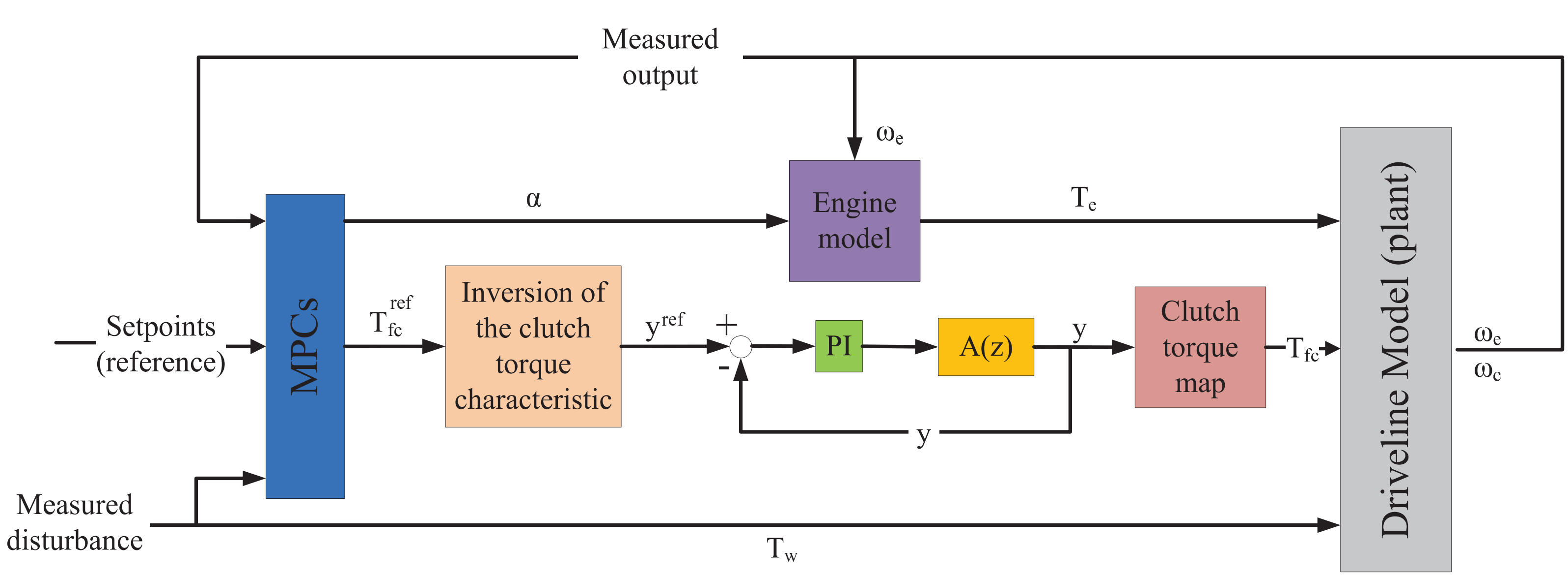

0.4
0.35
0.3
0.25
0.2
Friction Coefficient
0
0.5
1
1.5
2
2.5
3
v_s [m s^{-1}]
0.24 MPa (30 N)
0.48 MPa (60 N)

Fig. 4



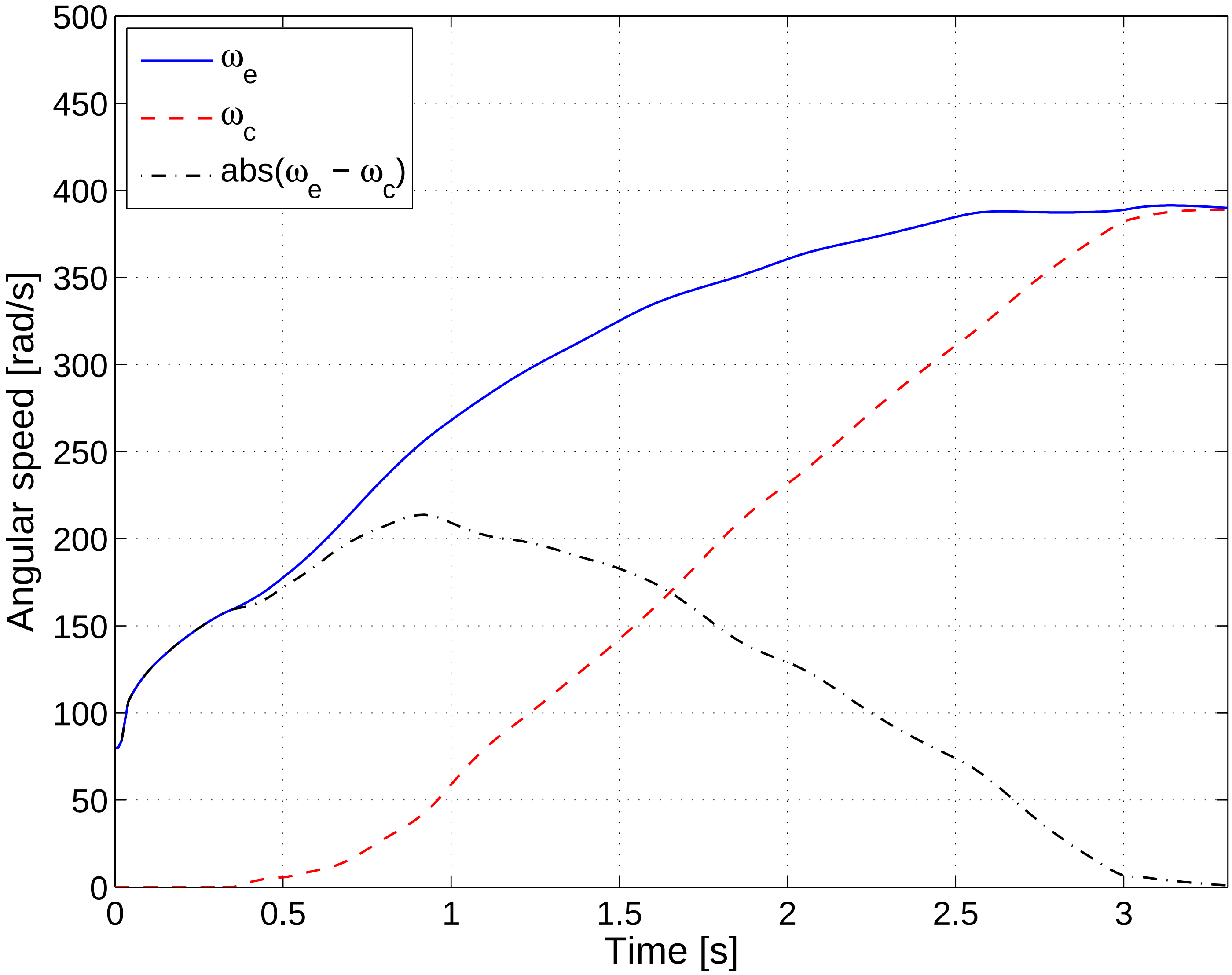
7000
6000
5000
4000
3000
2000
1000
0
3
3.5
4
4.5
5
5.5
6
6.5
7
7.5
8
Cushion spring reaction [N]
Bearing position [mm]
20°C
80°C
195°C
250°C
300°C

Fig. 5



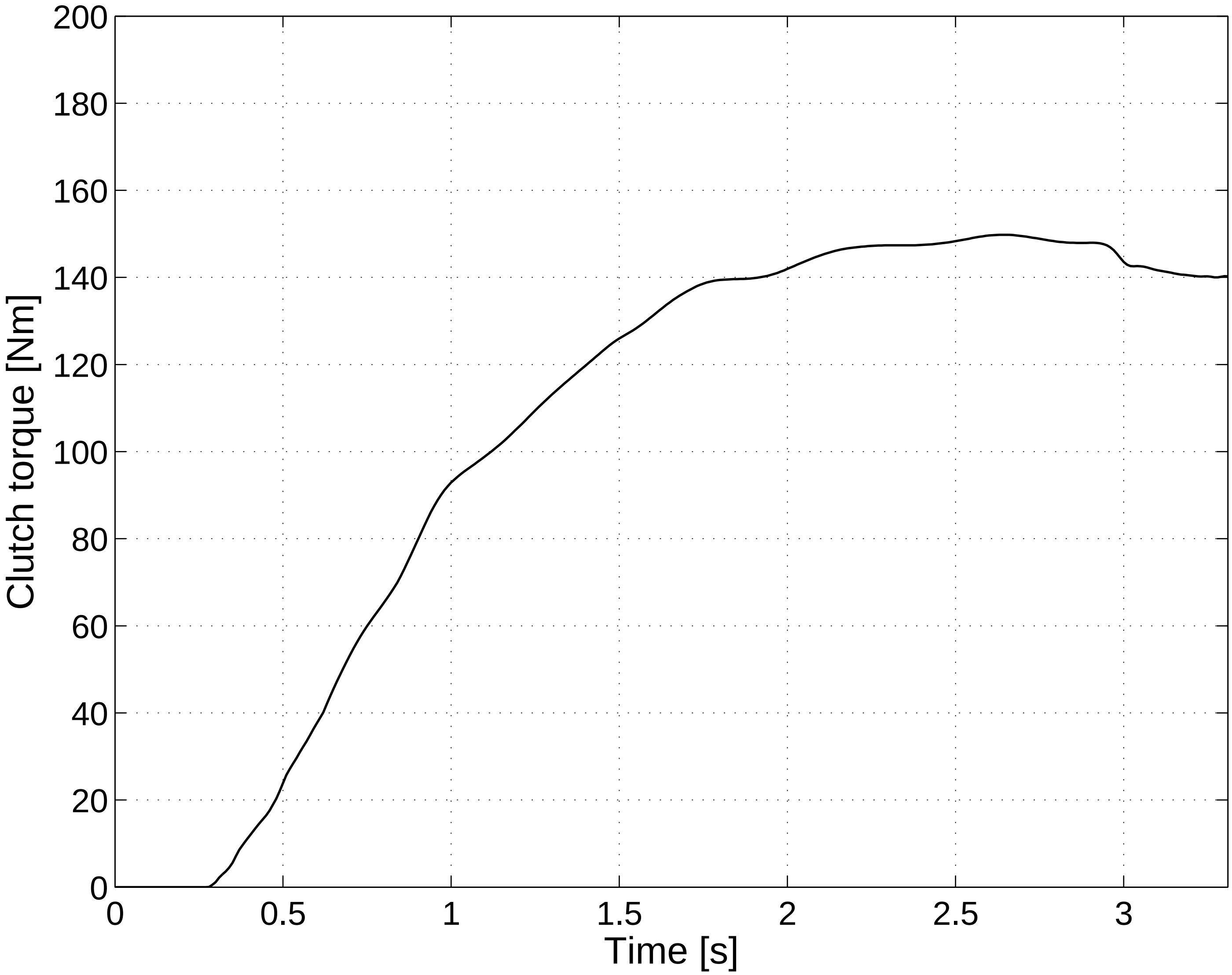
Measured
output
ωe
α
Engine
model
Te
Driveline Model (plant)
MPCs
Setpoints
(reference)
Tfc ref
Inversion of
the clutch
torque
characteristic
y ref
+
-
PI
A(z)
y
Clutch
torque
map
Tfc
ωe
ωc
y
Measured
disturbance
Tw

Fig. 6



500
450
400
350
300
250
200
150
100
50
0
0
0.5
1
1.5
2
2.5
3
Angular speed [rad/s]
Time [s]
ω_e
ω_c
abs($\omega_e - \omega_c$)

Fig. 7



Clutch torque [Nm]
200
180
160
140
120
100
80
60
40
20
0
0
0.5
1
1.5
2
2.5
3
Time [s]

Fig. 8

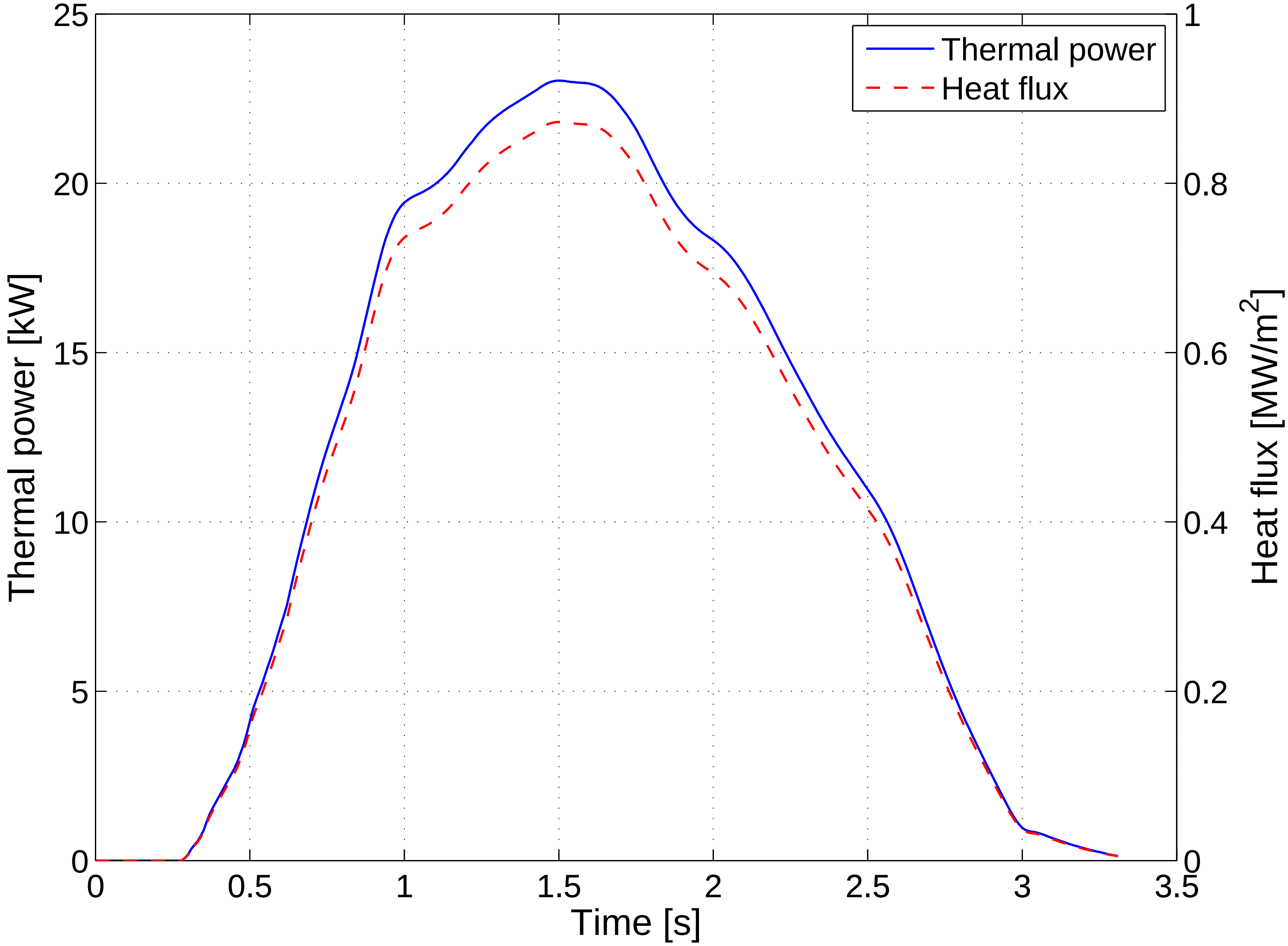
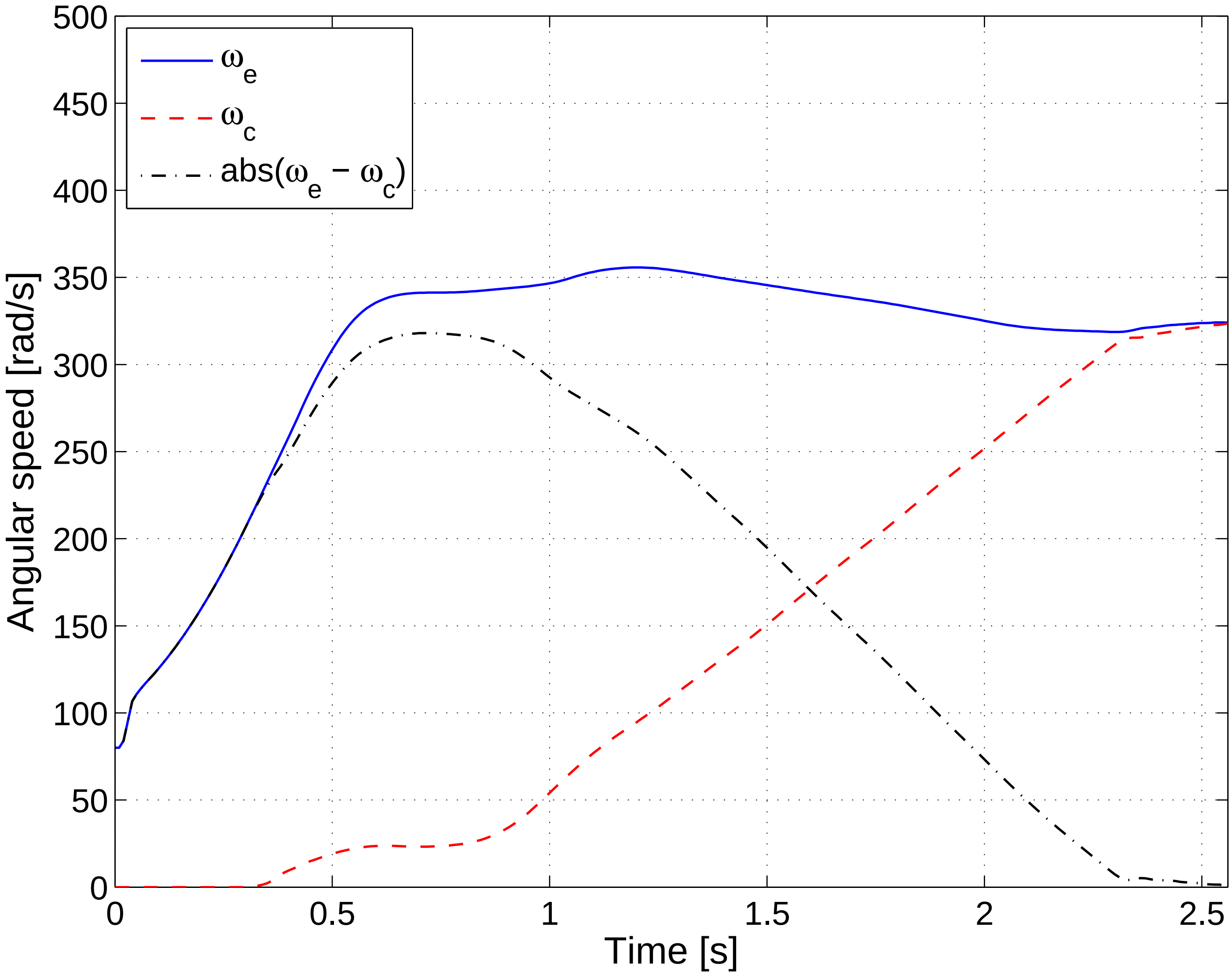

Thermal power
Heat flux
Thermal power [kW]
Heat flux [MW/m²]
Time [s]
0
0.5
1
1.5
2
2.5
3
3.5
0
5
10
15
20
25
0
0.2
0.4
0.6
0.8
1

Fig. 9



500
450
400
350
300
250
200
150
100
50
0
Angular speed [rad/s]
0
0.5
1
1.5
2
2.5
Time [s]
ω_e
ω_c
abs(ω_e − ω_c)

Fig. 10

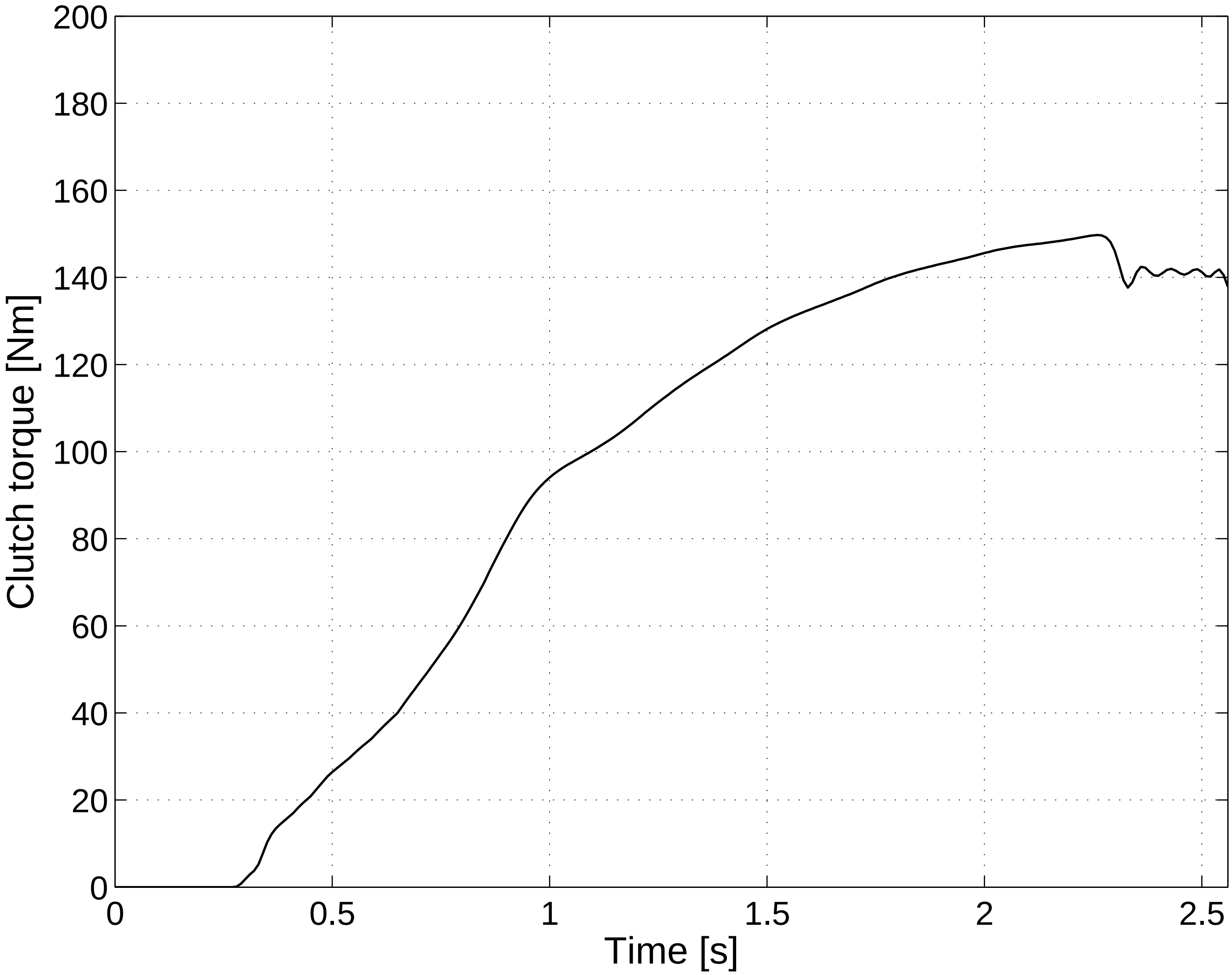
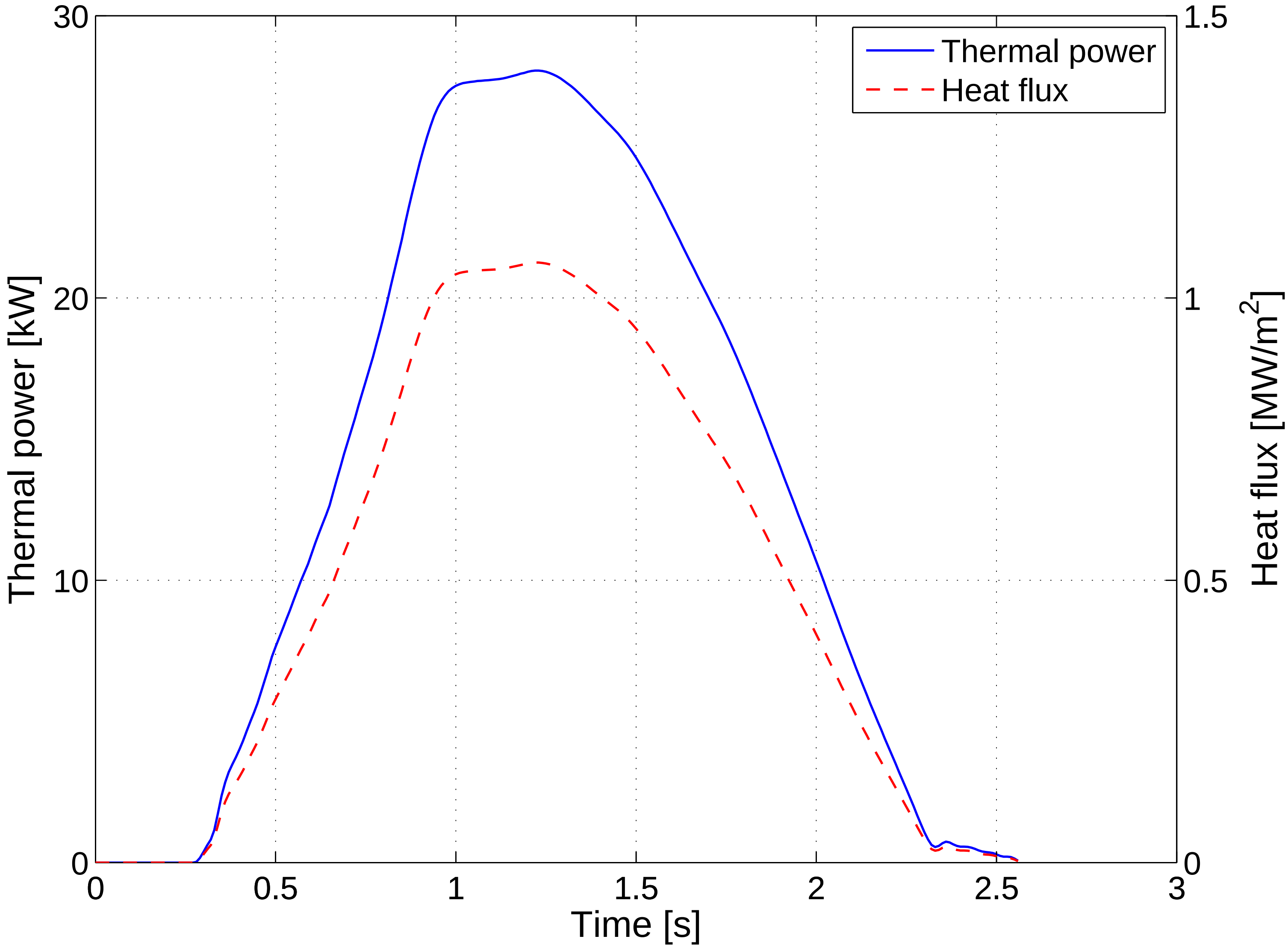
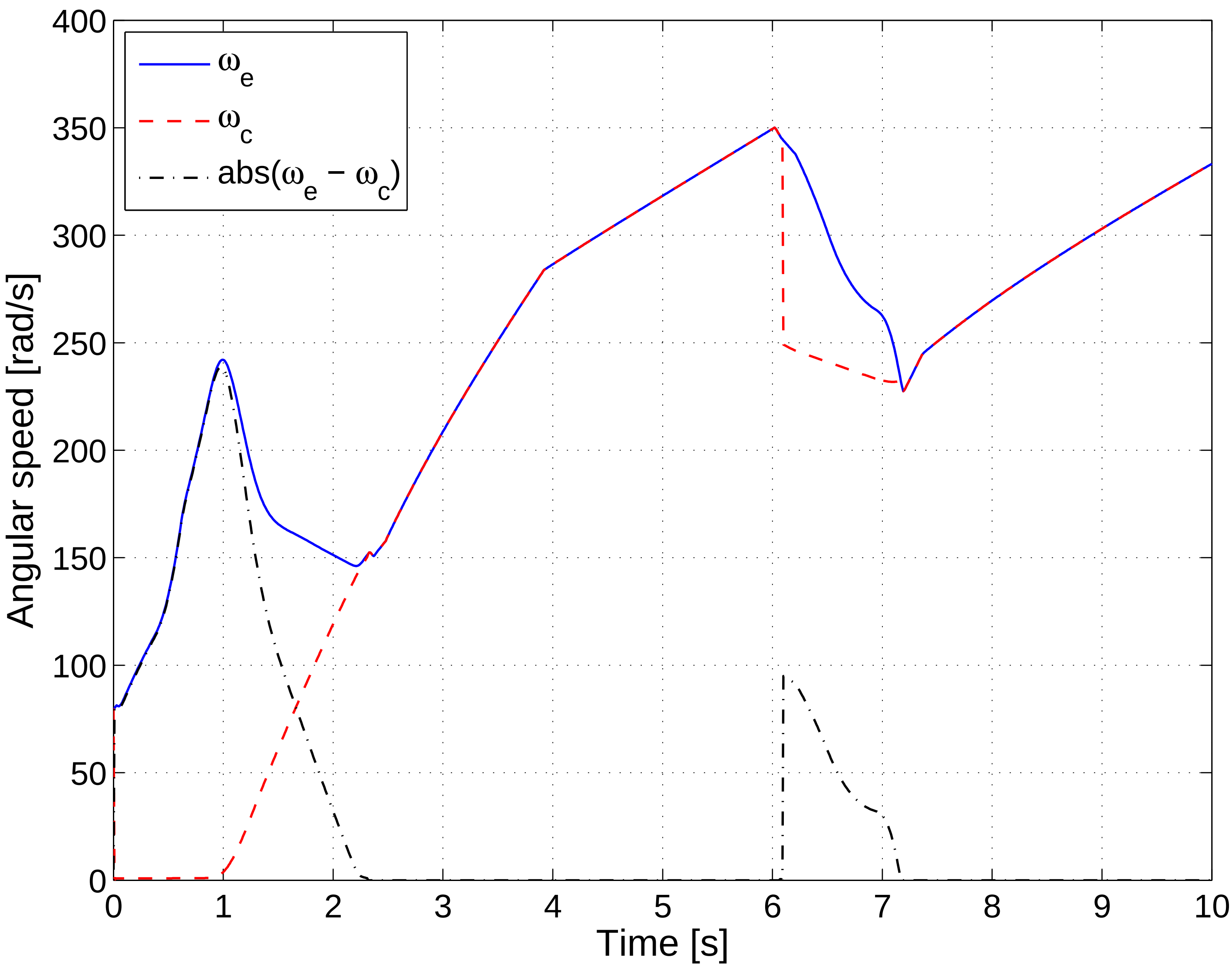

200
180
160
140
120
100
80
60
40
20
0
Clutch torque [Nm]
0
0.5
1
1.5
2
2.5
Time [s]

Fig. 11



Thermal power
Heat flux
Thermal power [kW]
Heat flux [MW/m2]
Time [s]
0
10
20
30
0
0.5
1
1.5
0
0.5
1
1.5
2
2.5
3

Fig. 12



ωe
ωc
abs(ωe − ωc)
Angular speed [rad/s]
Time [s]
0
50
100
150
200
250
300
350
400
0
1
2
3
4
5
6
7
8
9
10

Fig. 13

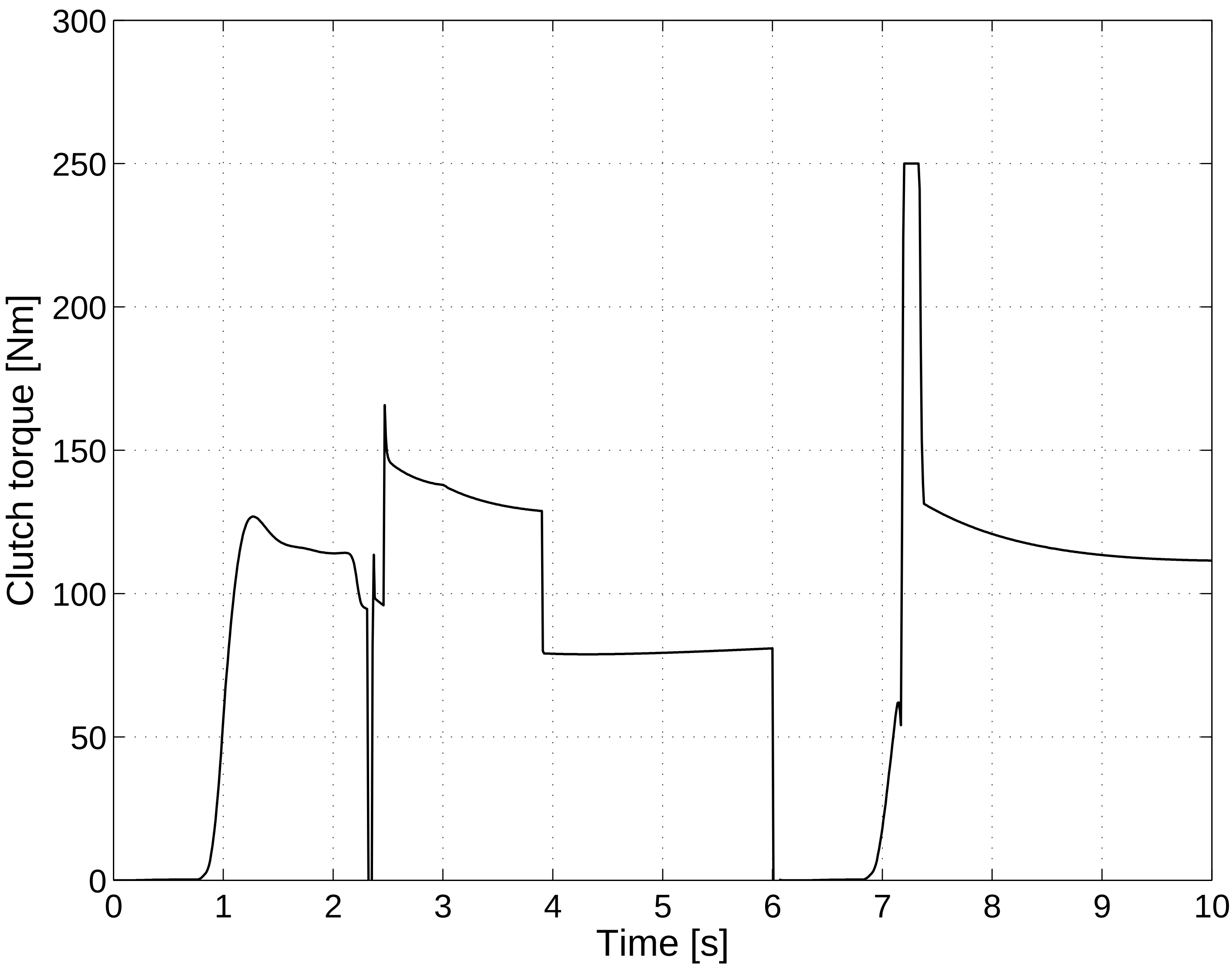


Fig. 14

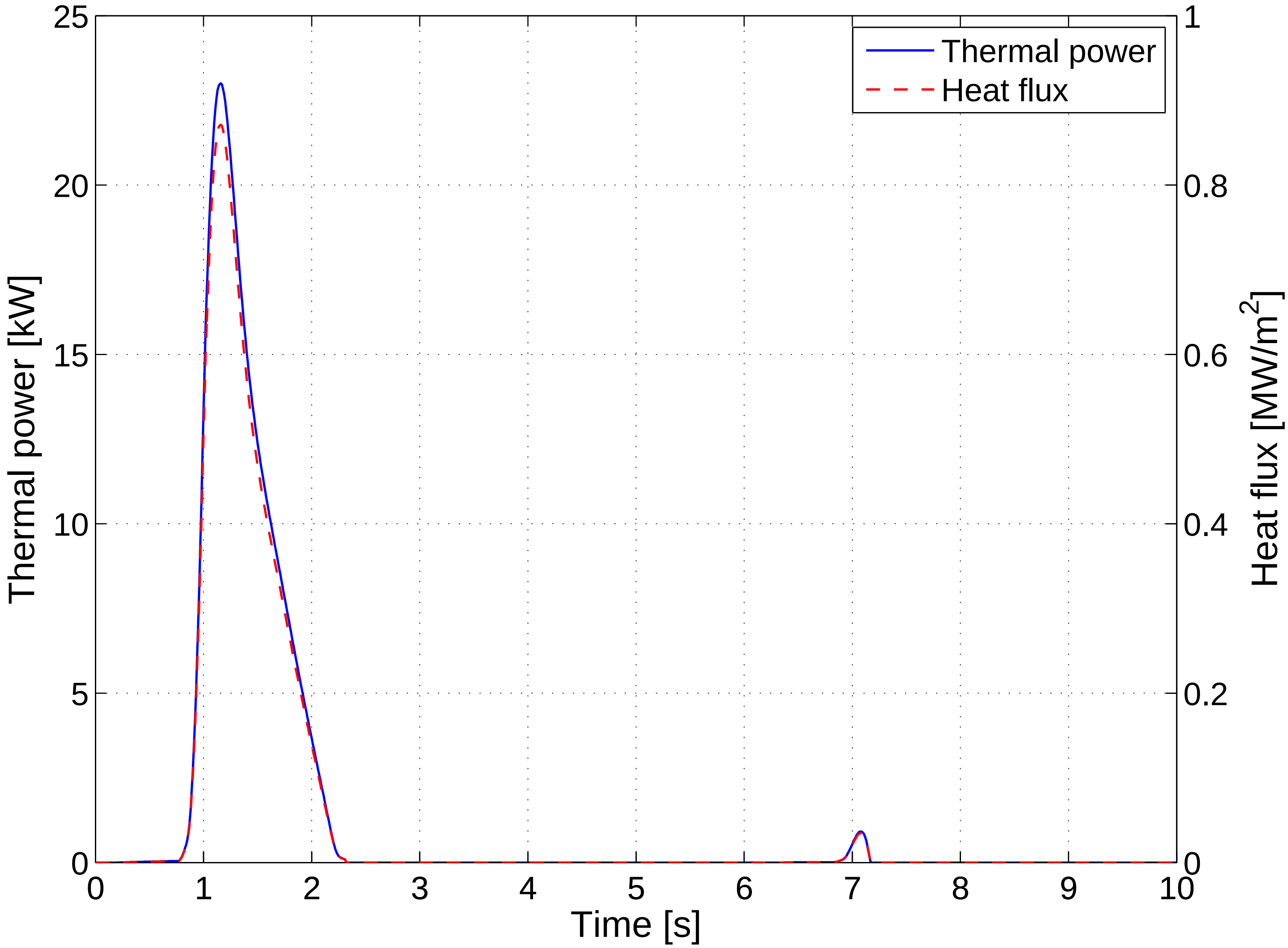
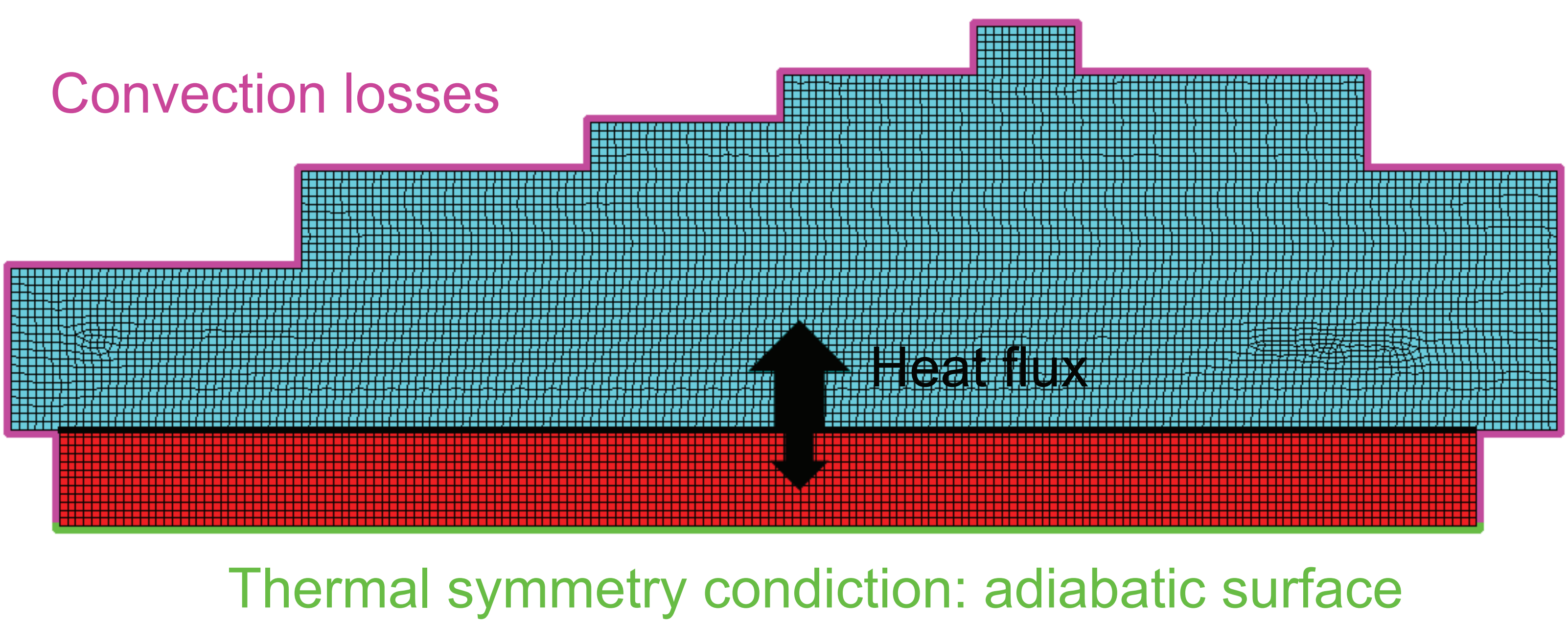

Thermal power
Heat flux
Thermal power [kW]
Heat flux [MW/m2]
Time [s]
0
5
10
15
20
25
0
0.2
0.4
0.6
0.8
1
0
1
2
3
4
5
6
7
8
9
10

Fig. 15



Convection losses
Heat flux
Thermal symmetry condiction: adiabatic surface

Fig. 16

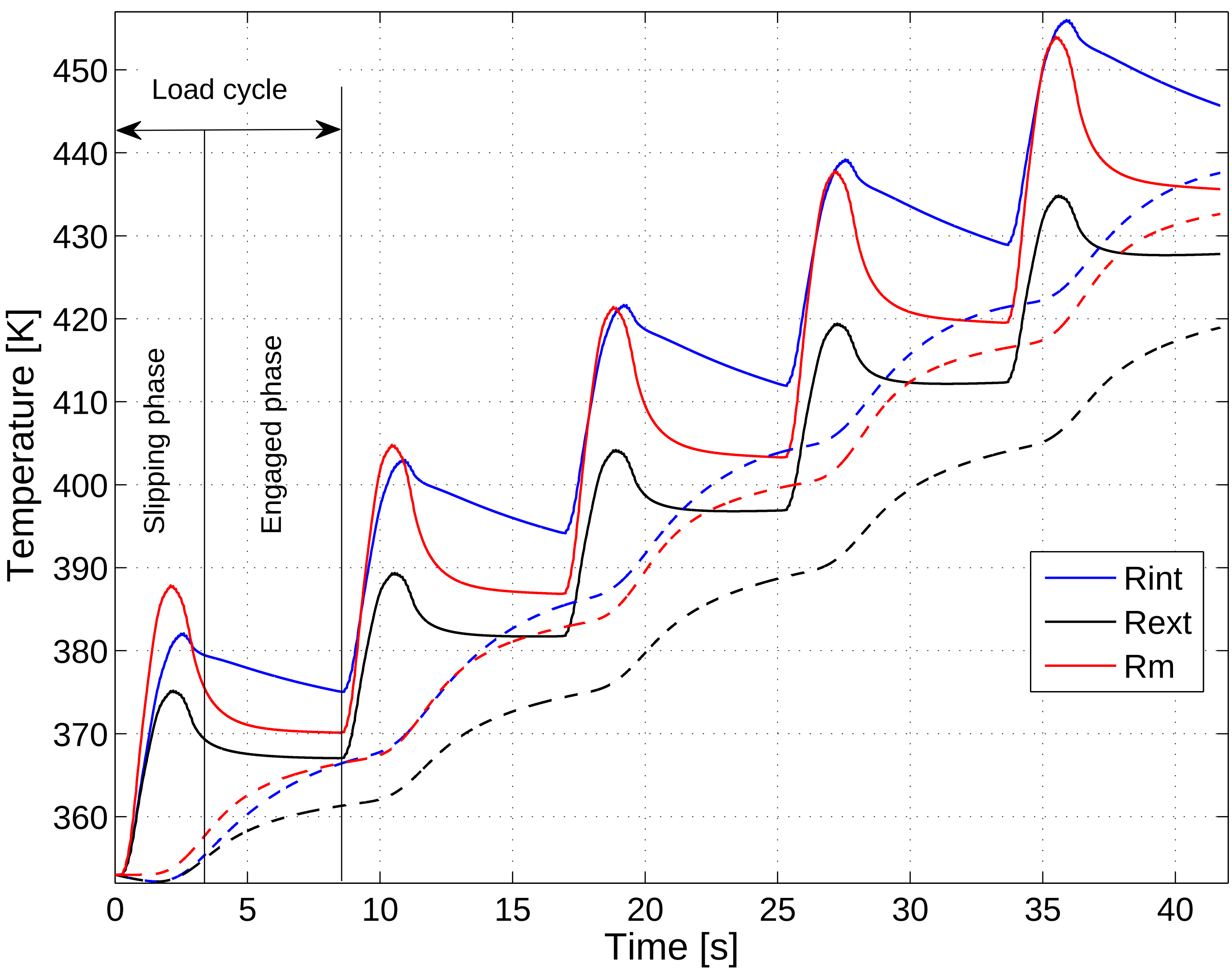

Load cycle
Slipping phase
Engaged phase
Temperature [K]
Time [s]
Rint
Rext
Rm
450
440
430
420
410
400
390
380
370
360
0
5
10
15
20
25
30
35
40

Fig. 17

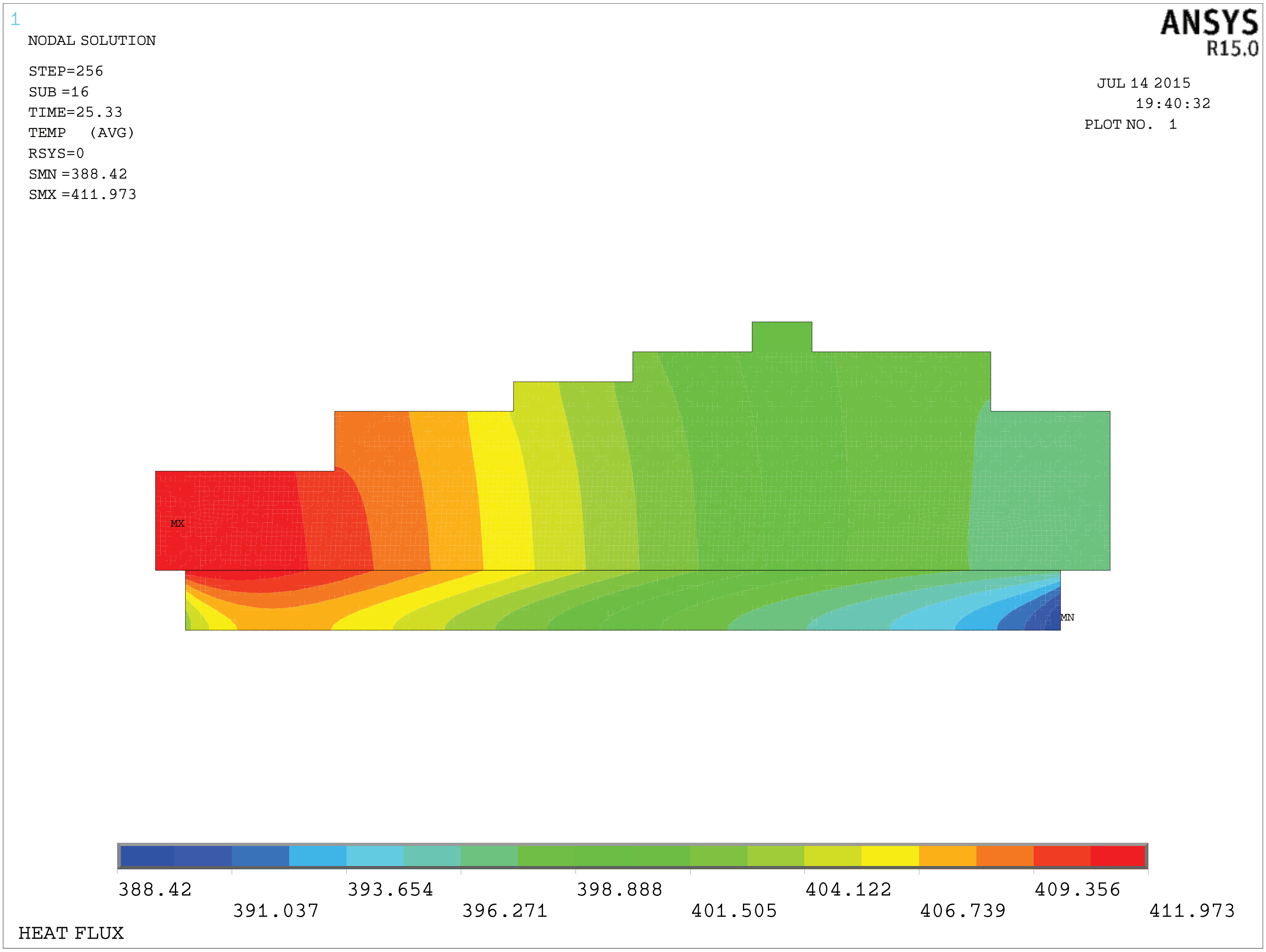

1
NODAL SOLUTION
STEP=256
SUB =16
TIME=25.33
TEMP (AVG)
RSYS=0
SMN =388.42
SMX =411.973
ANSYS
R15.0
JUL 14 2015
19:40:32
PLOT NO. 1
MX
MN
388.42
391.037
393.654
396.271
398.888
401.505
404.122
406.739
409.356
411.973
HEAT FLUX

Fig. 18

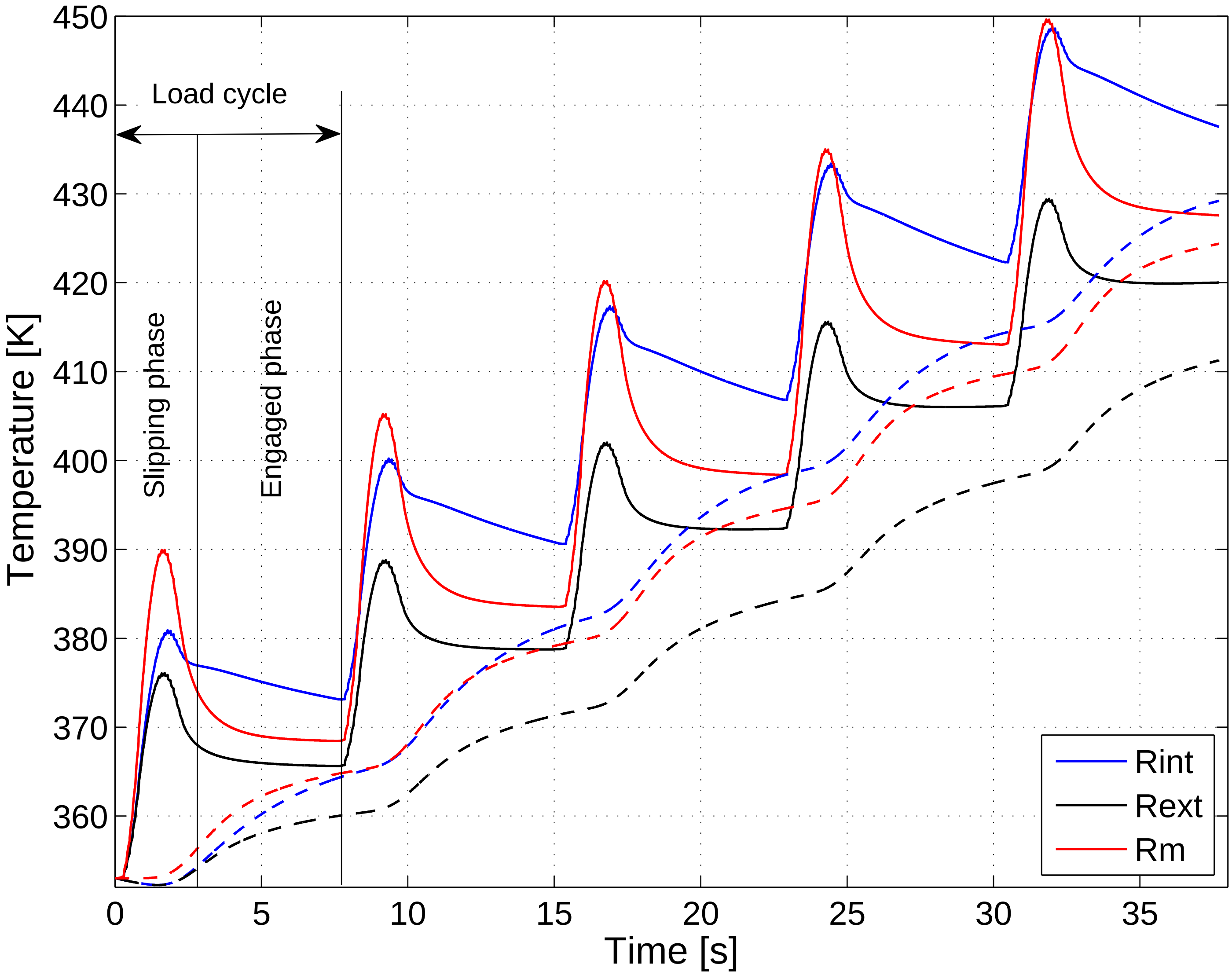
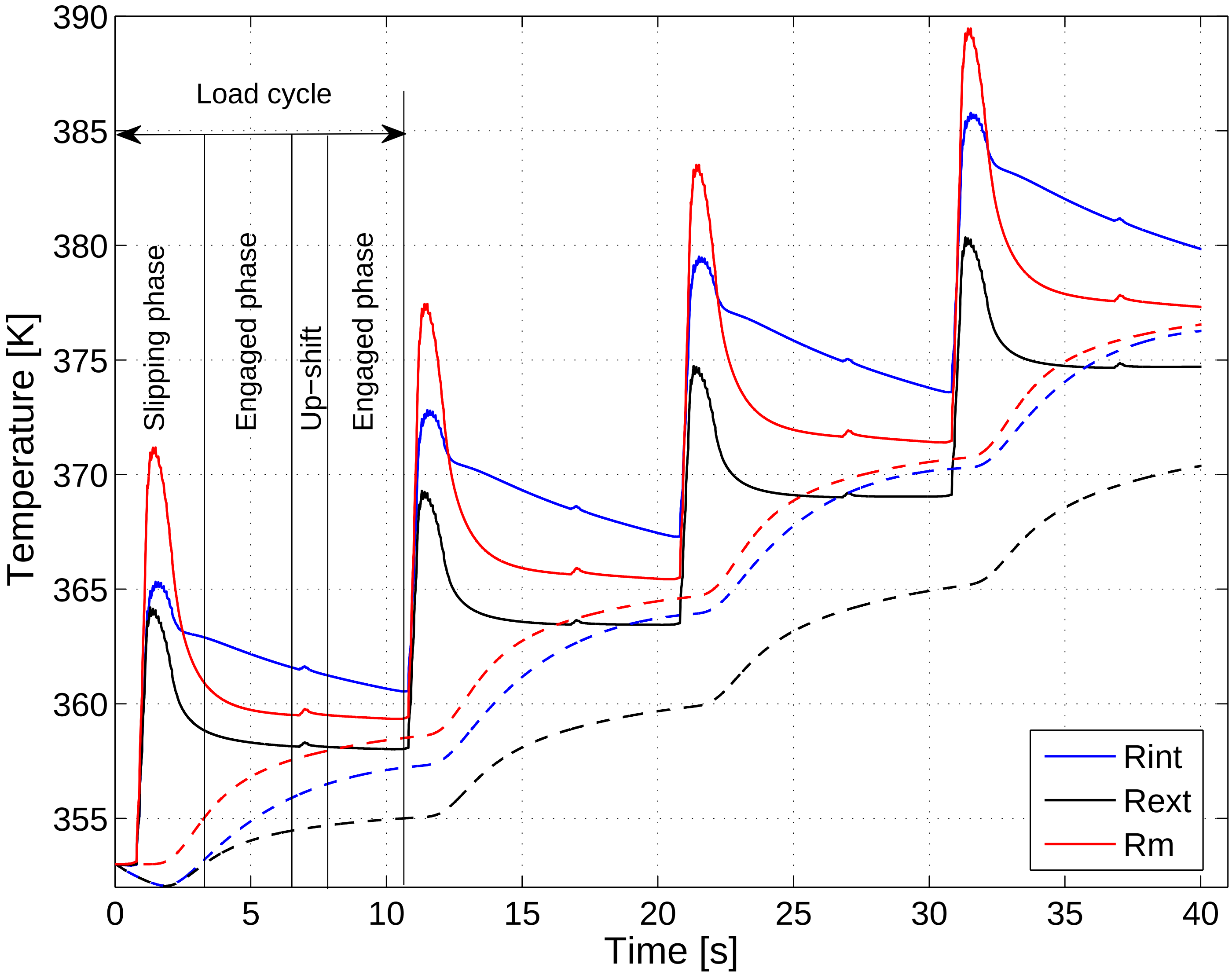

Load cycle
Slipping phase
Engaged phase
Temperature [K]
450
440
430
420
410
400
390
380
370
360
0
5
10
15
20
25
30
35
Time [s]
Rint
Rext
Rm

Fig. 19



Load cycle
Slipping phase
Engaged phase
Up−shift
Engaged phase
Temperature [K]
Time [s]
Rint
Rext
Rm
390
385
380
375
370
365
360
355
0
5
10
15
20
25
30
35
40